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(POPP'S SYSTEM)

BY

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## EXPERIMENTS UPON THE TRANSMISSION OF POWER BY COMPRESSED AIR IN PARIS.

(POPP'S SYSTEM.)

BY

**ALEXANDER B. W. KENNEDY,**

F.R.S., M.INST.C.E., EMERITUS PROFESSOR OF ENGINEERING
IN UNIVERSITY COLLEGE, LONDON.

## The Transmission and Distribution of Power from Central Stations by Compressed Air.

BY

**WILLIAM CAWTHORNE UNWIN,**

B.SC., F.R.S., M.INST.C.E.

EDITED BY

**F. E. IDELL, M.E.**

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# PREFACE.

THE transmission of power by compressed air has, in this country, been chiefly confined to tunnelling and mining operations; and while some of the plants for the generation of compressed air are large, no single plant is of such capacity as that in the Central Station of the Paris Compressed Air Co., where there may now be 10,000 horse-power generating compressed air.

A plant is to be erected at Niagara, N. Y., by the Niagara Power Co. There will be a central station for the generation of 5,000 h.p. by compressed air, and another for the generation of 5,000 h.p. by electricity, with the possible extension of either one of these to the amount of 100,000 horse-power. This power will be distributed, as wanted, to the factories on the lands of the company; then to an area within a two-mile radius. The radius will be increased gradually, until an area within a twenty-mile radius is covered. The deductions in Prof. Unwin's paper prove that power can be transmitted this distance without excessive loss of pressure, for he computes that air compressed to an initial pressure of 132 pounds per square inch can be carried a distance of 20 miles in a 30-inch main with a loss of pressure of only 12 per cent, and that from 40 to 50 per cent of the compressing power can be recovered in the motors if the air is used cold, and from 59 to 73 per cent if the air is reheated.

The following papers give the data of and deductions from experiments made by eminent authorities on the Paris system, and are presented to the reader *in extenso.*

F. E. I.

# EXPERIMENTS UPON THE TRANSMISSION OF POWER BY COMPRESSED AIR IN PARIS (POPP'S SYSTEM).

BY PROF. ALEXANDER B. W. KENNEDY,
F.R.S., M. INST. C. E.

THE subject of transmission of power to a distance is one of such immense importance in an industrial country like our own, that I am sure I need offer no apology for bringing this paper before you. In a case such as that which I have to describe, where the distance to which the power is to be transmitted is great enough to be measured in miles, the possible methods of transmission are not very numerous. Steam, water, air, and electricity seem to be the only four agents practically available. All four have been used to a greater or less extent, and with more or less complete success. In our own country the distribution and use of high-pressure water has been carried out with the greatest engineering skill and with correspondingly great success by the Hydraulic Power Company in London, and also in Hull. Electric transmission (for traction at least) has been at work on a small scale for some time in various places, and is about to be tried on a much larger scale on the Southwark Subway. In America it has been very widely used for traction on tramways, and on the Continent it has also been used to a certain extent for power transmission for general purposes. Steam has been used on a large scale in New York, but as yet its success does not seem to be unquestionable. Compressed air has, of course, been used over and over again in rough and uneconomical fashion in connection with tunnelling and boring work, but I think only two practical attempts have been made to utilize it economically and on a large scale for industrial purposes. Of these two, one has been made in Birmingham and the other in Paris. The *Birmingham* Compressed Air Power

Company has established works on a very large scale, but various causes have unfortunately combined to cause delay in the commencement of its operations, which indeed are hardly yet fairly started. In Paris, however, the transmission of power by compressed air has been in operation on a somewhat large scale and with very great mechanical success for a few years past. I have recently had occasion to spend some weeks in making experiments in connection with the Paris compressed-air plant, and have been given the fullest permission to publish the results of my experiments. To avoid any misunderstanding, I must premise that it is not my intention to institute any comparison between the different methods of power-transmission which I have mentioned. Such a comparison, to be of any value, would require for itself a paper at least as long as the one which I have to put before you, and is, therefore, in the nature of things impossible here. On this matter I wish to say only one thing, in view of a recent discussion on the Hydraulic Power Company's work in London, in which some comparisons were made between power transmission by air and by water. So far as I see, the two systems at present practically occupy different fields, and overlap but little. The work that each appears to do best is exactly that for which the other is least fitted. I see as little chance for air, just now, taking the place of water for lifts or cranes, as I do for water coming into common use in the driving of motors. I think it would be a pity if there were to be any impression that two systems were antagonistic which, in point of fact, rather supplement each other. Having said this much, I will limit myself to a description of the plant and methods used in Paris, and to a statement of the actual results obtained there, as determined by my own experiments on the spot. The plant and methods are by no means absolutely perfect; they are not only susceptible *of*, but are now receiving, considerable improvements in detail in the extensions which are being carried out. In what I have to say, however, I shall confine myself entirely to results actually obtained with the present plant, as it was working when I tested it two months ago, not giving it credit for the result of any of the improvements which have not yet been introduced throughout into the system of working. I shall only, after having given this statement of facts, state briefly my views as to the probable practical value of improvements which may be, or are being, carried out.

The work now carried on by the Paris Compressed Air Company has developed from very small beginnings, at first slowly, lately very fast. It originated in a pneumatic clock system, which was started about 1870 with a small "central station" in the Rue St. Anne, in the centre of Paris. This business grew gradually until it became far too large to be carried on from such a position, and a few years since a central station, with much enlarged machinery, was established in the Rue St. Fargeau, which is in Belleville, about 4½ miles east of the Madeleine. There are now about 8000 pneumatic clocks, public and private, in Paris, driven from St. Fargeau, and regulated by a standard clock in the Rue St. Anne; but as this part of the work, although it originally formed the basis of the whole system, is now a comparatively very small part of it, and is of an entirely special nature, I do not propose to say anything further about it here. Until about two years since, a pair of single-cylinder horizontal engines by Farcot, and a beam engine by Casse, sufficed for the whole work; but by that time the demand for compressed air for working motors had so increased that extension had become imperative, and the present working plant of six compound condensing engines, each working two air-compressors, with the necessary complement of boilers, was put down. This plant, except the compressors, was supplied from England by Messrs. Davey, Paxman & Co. of

Colchester. The compressors for the English engines were made in Switzerland on the Blanchod system. The demand for power is at present so great that, at certain hours of the day, practically the whole plant, old and new, indicating considerably over 2000 horsepower, is fully at work, and in consequence a duplicate main is being laid throughout, and new engines and compressors, half of them constructed by Davey, Paxman & Co., and half by John Cockerill & Co. of Seraing, are being pushed forward as rapidly as possible.

The general system of working is a follows: Fig. 1. The steam-cylinders (I) compress the air to a pressure of five atmospheres (six atmospheres absolute) or thereabouts in the compressor-cylinders (II). The air is drawn in direct from the engine-house, where I found it to be about 70 deg. Fahr., and after it has finally passed along the mains for some little distance it is again about the same temperature. It is therefore of the greatest importance to prevent its temperature rising during the compression, as all heat so taken up by the air represents work done in the steam-cylinders of which no part what-ever can be utilized. If the air were compressed adiabatically, i.e., without any cooling whatever, its temperature on leaving the compressor would be about 430 deg. Fahr., a temperature higher than that of saturated steam of 300 lbs. pressure per square inch. At St. Fargeau water for cooling is allowed to run into the cylinders through the suction-valve, during the suction stroke, in such quantity that the final temperature is only 150 deg. Fahr. So far the result is *satisfactory* enough, but owing, un-

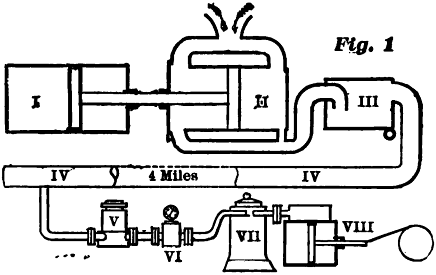

*Fig. 1*

fortunately, to the particular way in which the cooling water is utilized mechanically, the air does not get cooled until after it has been compressed, so that practically no benefit is obtained from the cooling in spite of the extent to which it occurs. The power expended, as we shall see presently, is practically equal to what would have been expended had the compression been adiabatic. The quantity of air dealt with at each revolution is 47.6 cubic feet (for the pair of double-acting compressing cylinders), which is equivalent to 3.55 lbs., the quantity of water used being about 2.4 lbs.

After compression, the air, now having an absolute pressure of six atmospheres and a temperature of 150 deg. Fahr., is pushed into large boiler-plate receivers (III) of which some are arranged to act as separators, and in these a large portion of the cooling water, which has been carried along mechanically by the air, is deposited and removed, before the air enters the mains (IV). The principal main is 300 mm. (11.8 in.) in diameter and about ⅜ in. thick. It is of cast-iron, made in lengths perfectly plain at each end, and connected by a very simple external joint made air-tight by india-rubber packing-rings. This joint leaves the pipe quite free endwise, and also allows all necessary sideway freedom, so that accidental distortion to a quite measurable extent is entirely without effect on the tightness of the joint. The mains are partly laid under roadways and footways, and partly slung from the roof of the sewer subways. They are supplied at intervals with automatic float traps for carrying off the entrained water and the water of saturation, as they deposit.

On entering a building on its way to a motor, the air is first passed through a meter (V) exactly as gas would be. The quantity passing is of course too great to allow anything like an ordinary gas-meter to be used; indeed, only inferential meters seem to have been at all successful. The meter actually in use in *Paris is a* small double cylindrical box.

The air passes by a branch through to the bottom of the inner box, up through it, down outside it between the two boxes, and away through a branch at the bottom opposite the inlet branch. The whole measuring apparatus is a little four- or six-armed fan, with aluminium or nickel vanes, placed near the bottom of the inner casing and communicating motion by a light vertical steel spindle to a clock-work register, like that of a gas-meter, placed on the top. The quantity recorded is simply the number of revolutions made by the fan, or some proportional number, and this is turned into cubic metres by multiplication by an arbitrary constant, determined by direct experiment. As to the working of this meter I shall have something to say later on; it is the only type used by the Paris company, and serves in a very large number of cases as basis of payment.

After passing the meter the air is carried through a reducing-valve (VI) by which the initial pressure in the motor is prevented from rising above a certain limit, which in practice appears to vary between 3½ and 5½ atmospheres absolute, according to the size of the motor in proportion to its work.

Between the reducing-valve (VI) and the motor (VIII) there is placed in all ordinary cases a small stove or heater (VII). This heater is simply a double cylindrical box of cast-iron, having an air-space between its outer and its inner walls. The air under pressure traverses this space, and is compelled, by suitably arranged baffle-plates, to circulate through it in such a fashion so as to come into contact with its whole surface. A coke fire is lighted in the interior of the stove, and the products of combustion are carried over the top of it, and made to pass downwards over its exterior surface, inside a sheet-iron casing, on their way to the chimney-flue. The heater for the motor on which I experimented (which indicated 10 to 12 horse-power) was about 21 in. in diameter and 2 ft. 9 in. high over all. The use of the heater, and the extent to which that use is advantageous and economical, are matters on which I shall touch later.

The motors used in Paris are mainly of two types. Up to one horse-power or thereabouts small rotary engines, of a form patented by M. Popp, are used; into the details of these it is not necessary to enter here. They start very readily, are easily governed, are provided with capital automatic lubricators worked by compressed air, run at a very high speed, and are altogether very convenient. They use the air with little or no expansion, without previous heating, and have, of course, no pretence to economy in use of air.

The larger sized motors, up to double-cylinder engines 12 in. by 14 in., which is the largest size used, are simply ordinary Davey-Paxman steam-engines, employed for air absolutely without any alteration or modification. These engines have, in most cases, automatic cut-off gear controlled by the governor, and can therefore easily work with the largest economical ratio of expansion for the not very high available initial pressure. I believe that in every case heaters are provided for these engines, although in some instances, where both power and refrigeration are required, they are used sparingly or not at all, in order to take advantage of the cooling due to expansion.

After this short description of the general arrangements of the Paris company's work I come now to the experiments which I made to ascertain its efficiency. Starting from the main engines at the central station, the particular matter which I had to determine was the indicated horse-power which would be shown by a small motor three or four miles from St. Fargeau for each indicated horse-power expended by the main engines on the air which passed through that motor. The ratio thus obtained would be the total indicated efficiency of the whole system of transmission. This ratio is in reality the *product* of a number of separate efficiencies, and the separate determination of these formed a necessary check on the value of the total efficiency. These separate efficiencies may be summarised as follows:

1. Mechanical efficiency of main engines, or ratio of work done in compressors to indicated work in steam-cylinders.
2. Efficiency of compressors, or ratio of maximum work which could be done in a motor by each cubic foot of compressed air at 70 deg. Fahr., to the work actually done in compressing that air.
3. Efficiency of mains, or ratio in which the capacity of the compressed air for doing work is reduced by friction and leakage.
4. Efficiency of reducing-valves, or ratio in which the capacity of the compressed air for doing work is reduced by the lowering of its initial pressure at the motor.
5. Indicated efficiency of motor, or ratio in which the actual indicated work done falls short of the maximum work

which the quantity of air measured through the meter could do after passing the reducing-valve.

The product of these five efficiencies is the total efficiency of transmission without the use of a heater. When a heater is used the matter is somewhat more complicated. All the ratios given above represent what may be called mechanical efficiencies, all of them have *unity* for their maximum attainable value. It is, therefore, not possible to introduce in direct combination with them a thermodynamic efficiency (ratio of additional heat supplied to additional work done), which has for its maximum value not unity, but 0.3 or some similar small value. This could only be done if the measurement of efficiency had started originally from the heat given to the steam instead of from the indicated horse-power, and this would have given numbers having a minimum of practical value or convenience. Probably the best practical measure of the efficiency of the whole transmission, when using heated air, is obtained by finding the equivalent in indicated horse-power at the central station of the coke used in the heater, and adding this to the indicated horse-power actually used. It would not be possible, by the expenditure of this or any other amount of indicated horse-power at the central station, to obtain the same results as by heating the air just before entering the motor, but that, of course, does not affect the question before us.

The determination of the indicated horse-power of the main engines presented no difficulty. I measured it on one pair of engines at different speeds from 21 to 44 revolutions per minute. At 31.5 revolutions per minute it amounted to 254.9, and at all speeds it was approximately 8.1 indicated horse-power per minute per revolution. The mechanical efficiency was sensibly the same at all speeds, viz., 84.5 per cent., as given in the table. There was no method available for ascertaining to what extent the real quantity of air

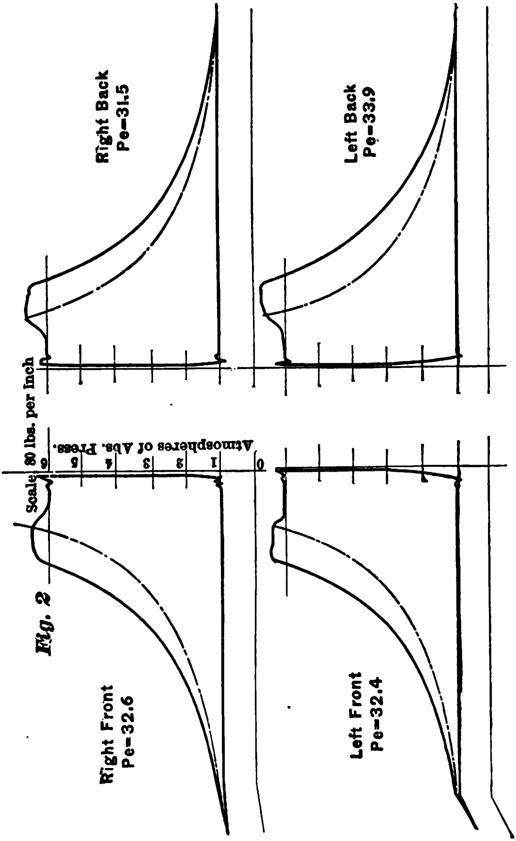
Scale 80 lbs. per Inch
Atmospheres of Abs. Press.
6
5
4
3
2
1
0
Right Back
Pe=31.5
Left Back
Pe=33.9
Right Front
Pe=32.6
Left Front
Pe=32.4

Fig. 2

delivered corresponded to the nominal volume swept through by the compressor-pistons. The indicator-diagrams, Fig. 2, show no signs of leakage past the valve, but there are no doubt various possible leakages which would not show on the diagrams. In the absence of any direct means of determination, however, I have assumed that the compressor-cylinders delivered their full volume, which corresponds to 348 cubic feet* of air per indicated horse-power per hour. This air has a weight of about 25 lbs. It may be pointed out that the water injected practically fills up the clearance-space at the end of each stroke.

I have already pointed out that at whatever temperature the air is delivered, it must fall to about its original temperature in the long length of mains before it reaches the motors. It is, therefore, a simple matter to find the maximum amount of work which can be done by the air delivered per indicated horse-power, for it simply amounts to the PV of the air at six atmospheres absolute and at 70 deg. Fahr., plus the work it can do in expanding adiabatically to a pressure of one atmosphere, and minus the work necessary to expel it from the cylinder at that pressure and at the corresponding temperature.† In the present case I find that this work is equivalent to 0.52 indicated horse-power for an hour, so that the efficiency of the compressors is 61 per cent. It will be seen from the diagrams that the compression curve falls only a little below the adiabatic, and as the area of the diagram is increased somewhat by the resistance of the discharge-valve, the work done is practically the same as if the compression had been adiabatic, and

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* Here and elsewhere, unless specially mentioned, volumes are supposed to be at atmospheric pressure and at 70 deg. Fahr., the actual admission temperature *during my* experiments.

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† In symbols, if suffixes 1 and 2 be used for the initial and final conditions of the air, if pressures be measured in pounds per square foot and volumes in cubic feet, the maximum work possible, without addition of heat, is

$$(P_1V_1 - P_2V_2)\frac{\gamma}{\gamma - 1} = 3.45\,(P_1V_1 - P_2V_2).$$

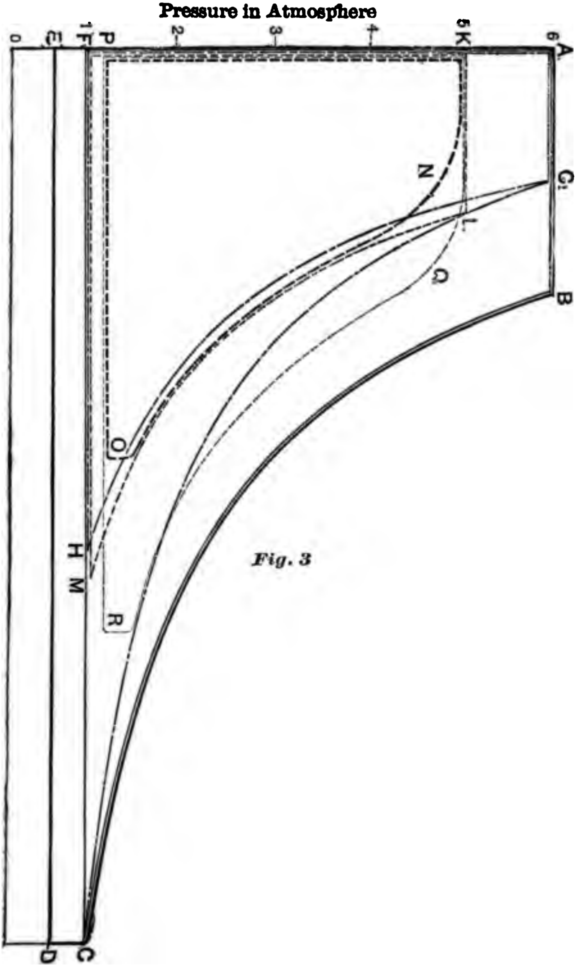


*Fig. 3*

there had been no cooling at all. This matter I have already referred to.

Fig. 3 shows graphically the relation between the quantities so far measured. If the area ABCDE represents on any scale the work done in the steam-cylinder, the area ABCF represents on the same scale the work done on the air in the compressors. CB is an adiabatic curve, CG an hyperbola, so that AG represents the volume of the compressed air when it has fallen in the mains to the temperature which it had initially at C. GH being again an adiabatic curve, the area AGHF (which is 61 per cent of ABCF, and 52 per cent of ABCDE) represents the maximum work which can be obtained from the air in a motor, the quantity calculated in the last paragraph.

If it be assumed that the temperature in the mains is constant, then any loss of pressure, due to friction, must be accompanied by an exactly equivalent increase of volume. Thus if (Fig. 3) the pressure falls from A to K, the volume *increases from* AG to KL, the point L lying on an isothermal through G. The loss of possible work due to such a reduction of pressure is represented by the difference between the areas AGHF and KLMF, in both of which the expansion curves are adiabatic. The change of temperature in passing through a reducing-valve is so small that it may be assumed without sensible error that the loss due to such a process may be calculated in the same fashion.

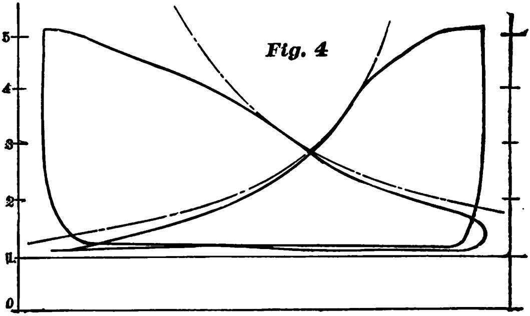
I determined the loss of head in the mains by a series of observations made simultaneously at known points in Paris and at St. Fargeau. The pressure-gauges used having been carefully compared, and all the necessary corrections made, I found the loss of pressure to vary from 0.35 to 0.25 atmosphere, according to the distance from St. Fargeau and the amount of air passing through the pipes. The average loss may be taken as 0.3 of an atmosphere at three miles from St. Fargeau, when the indicated horse-power there was about 1250 and the maximum velocity of the air in the mains about

1550 feet per minute. What proportion of this loss of head may have been due to leakage, and what the amount of leakage (if any) may have been, I had no means of determining. The duplicate main to which I have referred is not yet completed, and it was impossible to isolate any portion of the mains, even temporarily, to test for leakage. From the figures I have given, as well as from the nature of the pipe-joint, I cannot doubt that the leakage must, under any circumstances, have been extremely small.

In the table I have given approximate values of the loss due to fall of pressure in the mains and through the reducing valve, with various values of the total reduction of pressure. With a total reduction of half an atmosphere the combined efficiency of mains and valves is 0.96, reducing the maximum possible work at the motor to 0.5 indicated horse-power per indicated horse-power at the central station. Under these conditions the minimum possible consumption of air per indicated horse-power at the motor would be twice 348, or 696 cubic feet per hour.

The motor on which I made most of my experiments was an ordinary horizontal Davey-Paxman engine, with a single cylinder 8¼ inches in diameter and 12 inches stroke, fitted with automatic cut-off gear. For convenience sake I tested it at St. Fargeau and not in Paris, but I used a pressure only of 4½ atmospheres, which pressure I found to be exceeded on branch mains 3½ miles from St. Fargeau, where I made later experiments. The position of the motor did not, therefore, put it under any conditions different from those existing in the centre of Paris. I made a large number of experiments on this motor, under various conditions, individual experiments lasting from four hours down to half an hour. I shall here give figures representing only the four most important of my experiments, two with cold and two with heated air, averaging the two experiments in each case, so as not to burden my paper with unnecessary figures.

The motor, when indicating 9.9 horse-power, and making about 125 revolutions per minute, used 890 cubic feet of air per indicated horse-power per hour. (Its indicator-diagrams are shown in Fig. 4, their area being represented by KNO in Fig. 3.) The work which this quantity of air, at the given pressure and temperature, is theoretically capable of doing behind a piston, expanding down to atmospheric pressure, is equivalent to 1.27 horse-power for an hour. The indicated efficiency of the motor (the ratio expressing loss by rounding of curves, by insufficient expansion, by back pressure,



etc.) is therefore 0.79. This figure gives us a check on the ratios already worked out, for if they are right the air actually used should be $\frac{1}{0.79}$ times as great as the 696 cubic feet already allowed for. This would be 880 cubic feet, which represents of course a most satisfactory check. It will, however, be recognized that this agreement checks the figures only so far as they apply to air actually used, and would not be vitiated or in any way affected by losses by leakage.

The vital measurement of all the experiments was, of course, that of the quantity of air used. The air was passed through one of the fan-meters already described, readings of which were taken every quarter of an hour. After the experiments were over air was passed through the same meter at exactly the same pressure, and in as nearly as possible the same quantity, and then passed, at atmospheric pressure, through two large standard wet gas-meters. The readings of these were taken as correct, and the multiplier for the fan-meter determined from them. I found from numerous experiments on several fan-meters that this multiplier varied both with pressure and with quantity, but that the latter variation was very small within the limits of my experiments. I have no doubt that the air quantities which I shall give you are correct within about 2 per cent. plus or minus.

It will be seen that the total indicated efficiency of transmission, with cold air, is 0.39; in other words, that work requires to be done at the rate of 2.6 indicated horse-power at the central station per indicated horse-power at the motor. The motor was worked on a brake, and its mechanical efficiency was found to be 0.67, so that in round numbers 4 indicated horse-power were required at St. Fargeau per brake horse-power at the motor.

To examine the economy due to heating the air before using it, I used the same motor, working as nearly as possible *at the* same power and speed, and with exactly the same pressure, but passing the air between the meter and the engine through such a heating-stove as I have already described. I weighed all the coke used, and read the temperatures every five minutes during a four-hours' trial. The air was heated in passing through the stove, up to 315 degr. Fahr., with a consumption of about 0.39 lb. coke per indicated horse-power per hour. As the admission temperature on the cold trials was 83 deg. Fahr.* only, this corresponds to an increase of about 42 per cent in the volume of the air, and should therefore (had the indicated efficiency remained the same) have been accompanied by a decrease of air consumption in the ratio $\frac{1}{1.42}$ or 0.70. The air actually used was 665 cubic feet per indicated horse-power per hour, which is 0.75 of the 890 cubic feet formerly required, so that the full economy is very nearly realized. The indicator-diagrams are shown in Fig. 5, and are represented by KPQO in Fig. 3. An air consumption of 665 cubic feet per indicated horse-power per hour corresponds to an indicated efficiency over the whole system of 0.52, in

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* This somewhat high admission temperature was the only point in which the motor at St. Fargeau differed from those in Paris, where I found the admission temperature to be from 69 deg. to 71 deg. Fahr

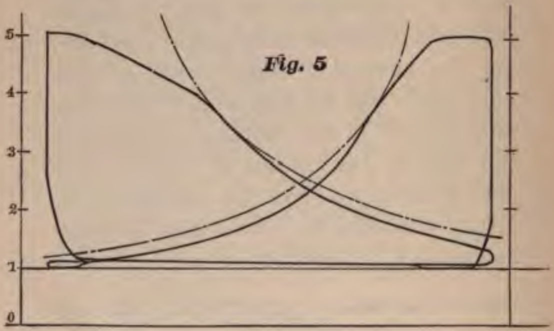
Fig. 5

other words 1.92 indicated horse-power is required at St. Fargeau per indicated horse-power at the motor. The mechanical efficiency of the motor was very much greater hot than cold, rising to 0.81. Hence about 2¼ indicated horse-power at St. Fargeau gave one brake horse-power at the motor. These figures, however, take no account of the coke burnt in the heater, and are, therefore, only to be considered as *apparent* efficiencies. Allowing for the value of the coke in the manner already described, the real indicated efficiency of the whole transmission is 0.47.

A shorter experiment with slightly higher temperatures and considerably larger indicated horse-power, gave still more economical results, the air consumption falling to 623 cubic feet per indicated horse-power per hour, an "apparent" indicated efficiency of 0.56. This experiment was not, however, of sufficient duration to allow of coke measurement.

As to the value of the preliminary heating, the figures which I have given show that it caused a saving of 225 cubic feet of air per indicated horse-power per hour, at a cost to the consumer of about 0.4 lb. of coke per indicated horse-power per hour. I do not doubt that the stoking of the heater during my experiment was much more careful than it would be in ordinary practice, although, on the other hand, it would not be difficult to design a more economical stove. If, however, the coke consumption were even doubled, it would only amount to 72 lbs. per day of nine hours for 10 indicated horse-power, the value of which might be 6d. or 7d. The air saved under the same circumstances would be over 20,000 cubic feet, the cost of which, at the high rate charged in Paris, would be 7s. 3d. There is no doubt therefore that to attain the maximum of economy the preliminary heating of the air should be carried as far as is practicable.

Of course heating the air serves the purpose also of preventing any chance of the exhaust-pipe becoming ice-clogged. I found this to happen once or twice when working with cold air, its occurrence depending rather on the amount of moisture in the air than on the exhaust temperature, for the engine, after running freely with an exhaust of −35 deg. Fahr., choked later on at +2 deg. Fahr. I do not think that in any case which I met with there would have been any trouble from choking had the exhaust-pipes been properly arranged. As it was, they were merely the ordinary vertical exhaust-pipes of a steam-engine, quite suitable for their original and for their intended purpose, but singularly unfitted for the purpose to which I was putting them.

Summarizing now the whole matter as regards efficiency, it may be said that the result of my detailed investigations is to show that the compressed-air transmission system in Paris is now being carried on on a large commercial scale in such a fashion that a small motor four miles away from the central station can indicate in round numbers 10 horse-power for 20 indicated horse-power at the station itself, allowing for the value of the coke used in heating the air, or for 25 indicated horse-power if the air be not heated at all. Larger motors than the one I tested (and there are a number of such in Paris) may work somewhat more, and smaller motors somewhat less, economically. The small rotary motors would, of course, be much less economical. The figures which I have given are, however, such as can be reached by any motor of between 5 and 25 indicated horse-power if worked at a fair power for its size. M. Victor Popp himself, and the engineers of the company, by no means content with the results already obtained, are experimenting in various directions with a view to greater economy, and I have not the least doubt that they will attain their end. But although I made several experiments on new apparatus, I prefer to leave their results here undiscussed, confining myself as strictly as possible to the work which has been already carried out and the economy of actual present working, rather than giving any credit for the result of improvements which, however certain they may be thought, are not yet actually carried out in practical work.

A system of transmission which has actually been carried out on a large commercial scale in such a way as to have an indicated efficiency of 50 per cent between prime mover and secondary motor, four or five miles apart, is one which needs no adventitious aids to commend it to notice, especially where its uses are so numerous and so varied, and its convenience so extremely great, as are those of compressed air. Both M. Victor Popp, who has organized and carried through the work of the Paris company, and Mr. James Paxman, who has designed and made the greater part of the machinery used, are to be heartily congratulated on the results which have attended their work.

While, however, I am unwilling to lay stress on possibilities which are not yet actualities, there can be no harm in saying that I have no doubt whatever that with mere improvement of existing methods and appliances, and without the adoption of any new or untried methods whatever, the new plant of the Paris company now being constructed can be made to have an indicated efficiency of 67 per cent instead of 50 per cent, and to give about 0.54 effective horse-power at the motor for each indicated horse-power at the central station, in the case of such a motor as that on which I experimented. Under these circumstances the air used per indicated horse-power at the motor would be 520 cubic feet, or 650 cubic feet per brake horse-power. I have the less hesitation in giving these hypothetical figures because the more important imperfections of M. Popp's transmission system arise from such a very obvious cause. Nothing, indeed, can be easier than to point out various weak points in the arrangements adopted, and yet, in spite of all such somewhat cheap criticism, the fact remains that no one has yet carried out a compressed-air transmission with anything approaching to the same success on anything like the same scale. The fact is that the success of the system has been essentially due rather to the practical good sense with which the work has been carried through than to any special novelty in the methods employed. The air-compressing arrangements at St. Fargeau are in no respect novel or specially perfect; they had been used over and over again before; there is no special advantage in M. Popp's rotary motor that may not probably be possessed by many other rotary motors; the larger motors are simply good ordinary steam-engines, such as can be bought any day in open market, used without the slightest alteration. Of the fan-meter it can only be said that it works well enough to allow progress to be made while it is being improved, and even of the coke stove one would not like to say very much more. The plan of heating compressed air before using it in a motor was first proposed many years ago. The great success which has attended the work in Paris has been attained because its directors have wisely chosen rather to set to work with imperfect apparatus, if only it were simple, fairly effective, and ready to hand, than to wait for the possible invention of novelties and improvements, or to risk the success of their start by the use of any unknown or untried apparatus, however promising its nature. They have had, moreover, the great advantage hitherto of being always asked for more air than they could supply, so that their works have grown and increased simply to meet a growing and increasing demand, and (fortunately perhaps) the urgency of the demand has left them no alternative but to meet it by the very simplest possible means.

I have already mentioned the great convenience and handiness which a compressed-air motor possesses. From the engineer's point of view these qualities are most striking. The engine starts, for instance, without the least hesitation even with full brake load on, directly the valve is opened, if the crank is just past the centre. This, of course, is impossible with a gas-engine, and hardly less impossible with any ordinary (single-cylinder) steam-engine. The absence of the heat and leakage, and of the noise and smell, which so often in greater or less degree accompany the smaller steam or gas motors, constitute a very much larger difference than could at first be thought possible. But from the consumer's point of view the advantages are even greater than from the engineer's. There is, first of all, the complete absence of danger and of nuisance of every kind. There is then the great saving of space, even as compared with a gas-engine, and much more as compared with a steam-engine and boiler. There is reduction of insurance on account of the entire absence of fire risk. Not only this, but the air motor seems to me completely to supply that most important industrial want, a motor suitable for "small industries," that is, for work carried on in workmen's own houses, or in very small workshops. For here it is not only mechanically most suitable, but in the nature of things it can be made to cool or ventilate, by its exhaust, to any desired extent. The sanitary advantage of this, in cases where work is carried on in confined spaces, can hardly be exaggerated. Even in a very large printing-office in Paris I found an almost unbearable atmosphere made quite pleasant as long as the motor was working, by allowing a portion of the exhaust to come into the room.

By using air direct from the mains in the motor, or by heating it only very slightly, the exhaust air can be, of course, so greatly reduced in temperature as to be available for freezing purposes. In one Paris restaurant, for instance, which I visited, I found that the exhaust was carried through a brick flue into the beer cellar. In this flue the carafes were set to freeze, and large moulds of block ice were also being made for table use, while the air was still cold enough in passing away through the beer cellar to render the use of ice for cooling quite unnecessary, even in the hottest weather. The nominal function of the engine in this case was the charging of batteries used in the electric lighting of the restaurant. The conjoint use of power and cold is common in Paris, the power being in this case generally applied in electric-lighting. While in any large city such as Paris, it is no doubt a great point that by a compressed-air system the handiest possible cooling appliances can be brought everywhere within reach, in tropical climates this is something rather of necessity than of luxury. In such cases we might have the apparent paradox of a motor worked essentially for its exhaust: the work done would be a bye-product; the cold air would be the principal thing. In such a case, if there were no useful work to be done, the motor could even be made (as has been suggested to me) to pump air back into the main, and thus virtually to about halve its air consumption. This possibility of "laying on" cold air in hot climates is, of course, a most important matter in connection with the future of compressed air.

## TABLE.

*Summary of Efficiencies of Compressed-air Transmission at Paris,* 1889, *between the Central Station at St. Fargeau and a* 10 *Horse-power Motor working with Pressure reduced to* 4½ *Atmospheres.*

(The figures below correspond to mean results of two experiments cold and two heated.)

| | |
|---|---|
| 1 indicated horse-power at Central Station gives 0.845 indicated horse-power in compressors, and corresponds to the compression of 348 cubic feet of air per hour from atmospheric pressure to 6 atmospheres absolute. (The weight of this air is about 25 lbs.). | Efficiency of main engines 0.845. |
| 0.845 indicated horse-power in compressors delivers as much air as will do 0.52 indicated horse-power in adiabatic expansion after it has fallen in temperature to the normal temperature of the mains. | Efficiency of compressors $\frac{0.52}{0.845} = 0.61$. |
| The fall of pressure in mains between Central Station and Paris (say 3 miles) reduces the possibility of work from 0.52 to 0.51 indicated horse-power. | Efficiency of transmission through mains $\frac{0.51}{0.52} = 0.98$ |
| The further fall of pressure through the reducing valve to 4½ atmospheres (5½ atmospheres absolute) reduces the possibility of work from 0.51 to 0.50. | Efficiency of reducing valve $\frac{0.50}{0.51} = 0.98$. |

The combined efficiency of the mains and reducing valve, between 5 and 4½ atmospheres, is thus 0.98 × 0.98 = 0.96. If the reduction had been to 4, 3½, or 3 atmospheres the corresponding efficiencies would have been 0.93, 0.89, and 0.85 respectively.

Incomplete expansion, wire-drawing, and other such causes reduce the actual indicated horse-power of the motor from 0.50 to 0.39.

Indicated efficiency of motor $\frac{0.39}{0.50}$ = 0.78.

Indicated efficiency of whole process with cold air 0.39.

By heating the air before it enters the motor to about 320 deg. Fahr., the actual indicated horse-power at the motor is, however, increased to 0.54. The ratio of gain by heating the air is, therefore, $\frac{0.54}{0.39} = 1.38$.

Apparent indicated efficiency of whole process with heated air, 0.54.

In this process additional heat is supplied by the combustion of about 0.39 lb. coke per indicated horse-power per hour, and if this be taken into account the real indicated efficiency of the whole process becomes 0.47 instead of 0.54.

Real indicated efficiency of whole process with heated air, 0.47.

Working with cold air the work spent in driving the motor itself reduces the available horse-power from 0.39 to 0.26.

Mechanical efficiency of motor, cold, 0.67.

Working with heated air the work spent in driving the motor itself reduces the available horse-power from 0.54 to 0.44.

Mechanical efficiency of motor, hot, 0.81.

Most of the compressed air in Paris is used for driving motors, but the work done by these is of the most varied kind. A list which I have gives the locality, use, and power of 225 installations, nearly all motors working at from ⅛ horse-power to 50 horse-power, all driven from St. Fargeau, and the great majority of them more than two miles away from it. In a number of cases the motor drives dynamo machines for electric lighting. In the offices of the *Figaro* and *Petit Journal* large motors are also used for printing, and there are many small printing establishments also worked by compressed air. Among the smaller industrial purposes for which the air motors are used in Paris, I find the driving of lathes for metal and wood, of circular saws, shearing-machines, drills, polishing-machines, and many others. They are used also in the workshops of carpenters, joiners and cabinet-makers, of smiths, of umbrella-makers, of collar-makers, of bookbinders, and naturally in a great many places where sewing-machines are used, both by dressmakers, tailors, and shoemakers, and from the smallest to the largest scale. They find application also in all sorts of industrial work, with confectioners, coffee-roasters, color-grinders, billiard-ball makers, in many departments of textile industry, and other matters too numerous to mention.

I may mention one particular instance of variety of application which interested me much. At the "Montagnes Russes" I found a large horizontal engine placed in a recess driving a dynamo and cells for the electric lighting of the whole building; a small vertical engine in another place worked the rotary pump, which actuated the "cascade;" two or three large air-driven fans in wooden shafts served for ventilation; and lastly, a simple connection on a flexible pipe threw the air-pressure into the beer-barrels as they were brought in, and transferred their contents to a height from which they could afterwards descend by gravity to the place where they were required.

As to the rate at which the compressed air is sold, I believe that in most of the larger installations the work is done by contract (at so much per lamp-hour in the case of the electric lighting), but in the smaller ones the air is charged for by meter. The rate of charge is 1.5 centimes per cubic metre (=8½ c. per 1000 cubic feet) for air used in motors and 2 centimes per cubic metre (= 11$\frac{3}{10}$c. per 1000 cubic feet) for air used directly. Air used for raising fluids is charged according to quantity raised. The former rate is equivalent to 4.25d. (= 8.5c.) per 1000 cubic feet (the volume in all cases being measured at atmospheric pressure). In addition to this, a fixed charge of from 100 to 250 francs (= $20.00 to $50.00) is made for pipes and connections, and a small rent for the motor, if it be not bought outright.

# THE TRANSMISSION AND DISTRIBUTION OF POWER FROM CENTRAL STATIONS BY COMPRESSED AIR.

By Prof. WILLIAM CAWTHORNE UNWIN, B. Sc., F.R.S., M. INST. C. E.

IN 1888 the author communicated to the Institution a short paper on "The Transmission of Power to great distances by Compressed Air." * In that paper the loss of pressure in air-mains and the efficiency of transmission were discussed, it is believed, with more theoretical completeness than had before been attempted. It was not within the scope of that paper to deal with the efficiencies of compressors and air-motors, and indeed at that time very little trustworthy

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* Minutes of Proceedings Inst. C.E., vol. xciii., p. 421.

experimental information was available as to the waste of work in air-machinery. However, for the purpose of examining the practical bearing of the general equations obtained, it was necessary to assume efficiencies for the compressors and air-motors, and some special cases of transmission of power were examined in detail. It may now be said that the assumptions made were not very inexact, for such systems of compressed air distribution as were in operation at that time. In 1889 Professor A. B. W. Kennedy made careful experiments on the system at work in Paris. He has given a summary at the end of his paper of all the losses of work which he found to occur in one case which was completely investigated. Air compressed at the central station, St. Fargeau, to six atmospheres (= 88.2 lbs. per square inch), was transmitted 5 kilometres (= 3.1 miles), reduced in pressure to four and a half atmospheres (= 66.15 lbs. per square inch), and used cold in an air motor of about 10 H.P. It was found that for 1 indicated H.P. expended in the steam-cylinder of the compressor, only 0.26 effective or brake H.P. was obtained at the air-motor. That is, the combined efficiency of compressor, main, and motor was 26 per cent. Now, in the author's paper it had been calculated that with initial compression to 58.8 lbs. per square inch absolute, and transmission a distance of a mile, the combined efficiency of compressor, main, and motor might range from 0.33 to 0.39 according to the size of main adopted. These results are rather better than that found by Professor Kennedy in the case which he examined. There is little doubt, however, that the air-motor experimented on by Professor Kennedy wasted power by leakage.

During the last year or two the systems of compressed-air distribution in Paris and Birmingham have been greatly extended, and similar plans for power-distribution in Dresden, Berlin, Vienna, Leeds and other towns have been proposed. Amongst the schemes submitted to the International Commission for the utilization of Niagara, there were three in which the power was to be distributed by compressed air. Hence the industrial importance of this system may now be considered as fully recognized.

Compressed air has often been used in transmitting power in mines and for tunnelling operations, where rough machinery is generally adopted, and where economy of power is not very important. It is not surprising that in many such cases extremely imperfect compressors and air-motors have been used, which, although they did the work required, wasted power extravagantly. It is from such experiences that an opinion has arisen that the efficiency of compressed-air transmission is very low. Better results have been obtained in the large systems for distributing power by compressed air in Paris and Birmingham. But it has been somewhat unfortunate that the types of compressor at first adopted were not such as to yield a high efficiency, and that many of the air-motors also were extremely bad. All the older compressors at Paris, and the large ones at Birmingham, are single-stage compressors, with very inadequate means of cooling the air during compression by water-jackets. At Birmingham the air is delivered into the mains at nearly 280° Fahrenheit, and practically the whole work expended in heating the air is wasted.

Now it has been known for a dozen years at least that spray injection into the cylinder is more effective in cooling the air than a water-jacket. There is also a much more powerful means of reducing the temperature of the air during compression. By compressing the air in two or more stages, and cooling it between the stages, the work expended in heating can be very greatly reduced. As early as 1881 the Norwalk Ironworks Company, Connecticut, U.S.A., constructed compound compressors, with an intercooler, with great advantage. Quite recently compound compressors have been made for the Paris system from the designs of Professor A. Riedler, of Berlin, with the result that two thirds to three fourths of the work expended in uselessly heating the air in the older machines in Paris is saved. At the same time investigations have been carried out on the air-motors in use in Paris, which show that in the construction of these also a considerable increase of efficiency can be secured.

In reconsidering the problem of distributing power by compressed air, especially in connection with the Niagara scheme, the author found nothing in the general method given in his previous paper which required revision. But the numerical results need correction in view of the improvements recently effected. Hardly any accurate experimental data were available two years ago. Now the very valuable treatise of Professor Riedler,* containing the results of observations carried out in Paris, furnishes a mass of trustworthy experimental results, both on the resistance of mains and the waste of work in compressors and air-motors. It appears worth while to re-examine the conditions of distribution of power by compressed air in the light of these new experiments.

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* Neue Erfahrungen über die Kraftversorgung von Paris durch Druckluft. Berlin, 1891.

## LOSS OF PRESSURE IN THE AIR-MAINS.

Hitherto information as to the resistance of air-mains has been scanty. The best experiments were those made by Mr. Stockalper on the pipes of the boring machines at the St. Gothard Tunnel. The new experiments carried out by Professor Riedler and Professor Gutermuth on the air-mains in Paris are therefore of great value. The Paris mains are larger than any hitherto tried, and by coupling up different mains at night, a length of 10¼ miles could be experimented on.

The main for the older Paris compressing-station consists of cast-iron pipes 11¾ inches or 0.98 foot in diameter. It was laid partly in the sewers, which involved the use of many bends. Part of the main in the Rue de Belleville is known to be leaky, and there are numerous draining-boxes, siphons, and stop-valves which cause resistance. In a more perfectly arranged main, no doubt, the loss of pressure would probably be somewhat less than in this old one in Paris. In the following investigation Professor Riedler's results are used, but the reductions from them and the conclusions deduced are the author's. The formula for the flow of air adopted is that given in the author's paper on "The Motion of Light Carriers in Pneumatic Tubes.*"

*Formula for the Flow of Air in Long Pipes.*—Let D be the diameter and L the length of a pipe in feet; $v$ the velocity of the fluid in feet per second; $\zeta$ the coefficient of friction, and H the head lost measured in feet of fluid under the given conditions. Then if the fluid is incompressible,

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* Minutes of Proceedings Inst. C.E., vol. xliii., 1876. Also article on Hydromechanics, Encyclopædia Britannica, Ninth Edition. Hydraulics, § 84.

$$\zeta \frac{v^2}{2g} = \frac{D}{4} \cdot \frac{H}{L};$$

$$H = \zeta \frac{v^2}{2g} \cdot \frac{4L}{D}.$$

This formula may be used for the flow of air in pipes, and indeed has been so used by Mr. Stockalper and others when the variations of pressure and density are small, so that mean values can be taken and the variations neglected. Professor Riedler uses this equation also, but with the artifice of dividing a long main into portions calculeatd separately—a method very cumbrous and not very accurate.

When air flows along a pipe there is necessarily a fall of pressure due to the resistance of the pipe, and consequently the volume and velocity of the air increases going along the pipe in the direction of motion. The effect of the resistance is to create eddying motions, which, as they subside, give back to the air the heat equivalent of the work expended in producing them. The result is that, apart from conduction to external bodies, the flow is isothermal. Generally, in compressed-air systems, the air is delivered into the mains at a temperature above that of the surrounding earth. The excess of heat is parted with by conduction, and the temperature falls to that of the ground, but no lower, for there can then be no further loss of heat by conduction.

Let P = the absolute pressure of the air in lbs. per square foot;
T = the absolute temperature;
G = the weight of the air per cubic foot in lbs.
V = the volume of the air per lb. in cubic feet.

Then

$$PV = \frac{P}{G} = cT, \quad . \quad . \quad (1)$$

where $c = 53.15$ for air. Taking the temperature of 60° Fahrenheit, so that $T = 461 + 60 = 521$,

$$cT = 53.15T = 27{,}690. \quad . \quad . \quad (2)$$

If air is flowing steadily in a pipe, the same weight of air flows across every transverse section per second. Hence, if W is the weight of air flowing per second, $\Omega$ the area of a cross-section, at which the velocity is $u$,

$$W = G\Omega u = \text{constant}; \quad . \quad (3)$$

combining (1) and (3),

$$\Omega u P = c T W. \quad . \quad . \quad (3a)$$

*Fig.* 1.

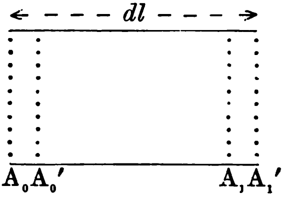

*Fig.* 1 represents a short length $dl$ of an air-main, between transverse sections $A_0$, $A_1$. Let $d$ be the diameter, $\Omega$ the cross-section, $m$ the hydraulic mean radius of the pipe. Let P and $u$ be the pressure and velocity at $A_0$, $P + dP$ and $u + du$ the same quantities at $A_1$. Let W be the weight of air flowing through the pipe per second. The units are feet and lbs.

If in a short time $dt$ the mass $A_0A_1$ comes to $A_0'A_1'$, then $A_0A_0' = u\,dt$ and $A_1A_1' = (u + du)\,dt$.

By analogy with liquids, the head lost in friction, measured in feet of fluid, is

$$\zeta \frac{u^2}{2g} \frac{dl}{m}.$$

Let $H = \dfrac{u^2}{2g}$; then the head lost is

$$\zeta H \frac{dl}{m};$$

and since $W\,dt$ is the flow through the space considered in the time $dt$, the work expended in friction is,

$$- \zeta \frac{H}{m} W\,dl\,dt.$$

The change of kinetic energy in the time $dt$ is the difference of the kinetic energy of $A_1A_1'$ and $A_0A_0'$; that is,

$$\frac{W dt}{2g} \{ (u + du)^2 - u^2 \}$$

$$= \frac{W}{g} u\,du\,dt = W\,dH\,dt.$$

The work of expansion of $\Omega u\, dt$ cubic feet of air to $\Omega(u + du)\, dt$, at a pressure initially P, is $\Omega \mathrm{P}\, du\, dt$. But from (3*a*)

$$u = \frac{c\mathrm{TW}}{\Omega \mathrm{P}},$$

$$\frac{du}{d\mathrm{P}} = -\frac{c\mathrm{TW}}{\Omega \mathrm{P}^2},$$

and the work done by expansion is

$$-\frac{c\mathrm{TW}}{\mathrm{P}} d\mathrm{P}\, dt.$$

The work done by gravity is zero if the pipe is horizontal, and in most cases may be neglected without great error. The work of the pressures on the sections $\mathrm{A}_0\mathrm{A}_1$ is

$$\mathrm{P}\Omega u\, dt - (\mathrm{P} + d\mathrm{P})\Omega(u + du)\, dt$$
$$= -(\mathrm{P}\, du + u\, d\mathrm{P})\Omega\, dt.$$

But if the temperature is constant,

$$\mathrm{P}u = \text{constant},$$
$$\mathrm{P}\, du + u\, d\mathrm{P} = 0,$$

and the work of the pressures is zero.

Adding together the quantities of work

and equating them to the change of kinetic energy,

$$W\,dH\,dt = -\frac{cTW}{P}dP\,dt - \zeta\frac{H}{m}W\,dl\,dt$$

$$dH + \frac{cT}{P}dP + \zeta\frac{H}{m}dl = 0,$$

$$\frac{dH}{H} + \frac{cT}{HP}dP + \zeta\frac{dl}{m} = 0. \quad . \quad (4)$$

But

$$u = \frac{cTW}{\Omega P},$$

$$H = \frac{u^2}{2g} = \frac{c^2T^2W^2}{2g\Omega^2P^2}.$$

$$\frac{dH}{H} + \frac{2g\Omega^2P}{cTW^2}dP + \zeta\frac{dl}{m} = 0.$$

For pipes of uniform section $\Omega$ and $m$ are constant, for steady motion W is constant, and for isothermal flow T is constant. Integrating,

$$\log H + \frac{g\Omega^2P^2}{W^2cT} + \zeta\frac{l}{m} = \text{constant}. \quad . \quad (5)$$

For $l = 0$, let $H = H_1$ and $P = P_1$,
" $l = L$, let $H = H_2$ and $P = P_2$,

$$\log\frac{H_2}{H_1} + \frac{g\Omega^2}{W^2cT}(P_2^2 - P_1^2) + \zeta\frac{L}{m} = 0, (6)$$

where $P_1$ is the greater and $P_2$ the less pressure. By replacing $H_1$, $H_2$, and W,

$$\log\frac{P_1}{P_2} + \frac{gcT}{u_1^2 P_1^2}(P_2^2 - P_1^2) + \zeta\frac{L}{m} = 0. \quad (6a)$$

Hence the initial velocity in the pipe is

$$u. = \sqrt{\left\{\frac{gcT(P_1^2 - P_2^2)}{P_1^2\left(\zeta\frac{L}{m} + \log\frac{P_1}{P_2}\right)}\right\}};$$

and when L is great, $\log\frac{P_1}{P_2}$ is comparatively small compared with the other term in the bracket. Then

$$u_1 = \sqrt{\left\{\frac{gcTm}{\zeta L}\,\frac{(P_1^2 - P_2^2)}{P_1^2}\right\}} \ldots (7)$$

For pipes of circular section $m = \frac{d'}{4}$, where $d'$ is the diameter in feet. Let $cT = 27{,}690$, and let $p_1$, $p_2$ be the pressures in lbs. per square inch. Then

$$u_1 = \sqrt{\left\{222{,}900\frac{d'}{\zeta L}\,\frac{(p_1^2 - p_2^2)}{p_1^2}\right\}}. \quad (7a)$$

This equation is easily used. In some cases the approximate equation

$$u_1 = \left(1.1319 - 0.7264\frac{p_1}{p_0}\right)\sqrt{\left(222{,}900\frac{d'}{\zeta L},\right.} \qquad (8)$$

may be more convenient.

If the terminal pressure $p_2$ is required in terms of the initial pressure $p_1$, then

$$p_2 = p_1\sqrt{\left\{1 - \frac{\zeta u_1^2 L}{222{,}900 d'}\right\}}. \qquad (9)$$

If from a series of experiments $\zeta$ is to be found,

$$\zeta = 222{,}900\frac{d'}{u_1^2 L}\,\frac{(p_1^2 - p_2^2)}{p_1^2}. \qquad (10)$$

Before proceeding to calculate the value of $\zeta$ from the experiments of Professor Riedler and Professor Gutermuth on the Paris mains, it is necessary to examine the corrections to be made for leakage and the special resistances of the draining-boxes.

*Loss of Air by Leakage from the Main.*—Special experiments were made by Professor Riedler and Professor Gutermuth to determine the leakage loss in portions of the old Paris main. These gave the following results:

| | I. Southern Section. Central Works to Pl. de la Concorde. | II. Entire Pipe Line. | III. Central Station to Pl. de la Concorde. | IV. Entire Pipe Line. | V. Northern Section. Central Station to Rue de Belleville. | VI. Rue des Pyrenées. |
|---|---|---|---|---|---|---|
| Length in miles... | 5.69 | 10.69 | 5.69 | 10.69 | 0.87 | 0.34 |
| Initial pressure in atmospheres.. | 6.5 | 6.9 | 7.0 | 6.7 | 6.0 | 6.1 |
| Terminal pressure. | 6.0 | 5.9 | 6.43 | 5.82 | 5.0 | 3.7 |
| Loss of pressure per hour in atmospheres.. | 1.5 | 1.5 | 0.57 | 1.32 | 0.6 | 0.56 |
| Loss of air in cu. ft. per hour, reckoned at atmospheric pressure. | 34300 | 67000 | 13000 | 59000 | 2100 | 782 |
| Loss of air in cubic feet per mile per hour. | 6030 | 6270 | 2285 | 5520 | 2414 | 2300 |

It is stated that during trials I, II, and IV considerable leakage was known to exist at the Central Station and the Rue de Belleville. In all the trials the air consumed by the clock system and some small motors not stopped is included. Reduced, as shown in the last line, the results are consistent if it is assumed that the leakage was really greater in the sections included in I, II, and IV than in the other sections.

Now, with an initial gauge-pressure of 6 atmospheres and an initial velocity of 30 feet per second, the main would deliver about 600,000 cubic feet of air per hour, reckoned at atmospheric pressure. Then the percentage of loss by leakage per mile per hour would be as follows:

| | I. | II. | III. | IV. | V. | VI. |
|---|---|---|---|---|---|---|
| Per cent of air lost by leakage per mile per hour.... | 1.00 | 1.05 | 0.38 | 0.91 | 0.40 | 0.38 |

Professor Riedler believes tha in newer and better-laid mains the leakage is considerably reduced. In any case it appears that the loss is small and practically can be neglected except in very long transmissions, at least when the main is delivering a full supply. It should be remembered, however, that the leakage is proportionately greatest in those hours when the pressure has to be maintained in the mains, but the demand for power is small. Where the demand for power is very variable this loss might become an appreciable factor. Looking to the uncertainty as to the amount and law of variation of leakage in different cases, it will be neglected in this investigation, taking the experiments above referred to as a proof that it is not a serious quantity even in the Paris installation, and that in newer mains laid with the experience now gained it would be still less important.

*Correction for Loss of Pressure at Draining-tanks.*—In some of the mains experimented on by Professor Riedler there were draining-tanks at which there was a sensible loss of pressure. The resistance at such a receiver must be analogous to the loss at a sudden enlargement of a pipe, and we may therefore write, if $\mathfrak{P}$ is the pressure lost, $p$ the absolute pressure of the air, and $v$ its velocity,

$$\mathfrak{P} = \zeta_d p v^2,$$

where $\zeta_d$ is the coefficient of resistance to be determined by experiment. Some experiments were made by Professor Riedler on these draining-tanks. The following table gives the results and the calculated value of $\zeta_d$:

| Absolute Pressure $p$. | Velocity of Air in Main. | Observed Loss of Pressure at One Draining-tank, $\mathfrak{P}$. | Calculated Value of $\zeta_d$. |
|---|---|---|---|
| Lbs. per Sq. Inch. | Feet per Second. | | |
| 104.6 | 19.5 | 0.94 | 0.00002363 |
| 104.1 | 19.0 | 0.95 | 0.00002527 |
| 84.1 | 28.5 | 2.42 | 0.00003543 |
| 106.9 | 24.25 | 2.14 | 0.00003404 |
| 102.0 | 18.25 | 1.00 | 0.00002943 |
| | | Mean . . . | 0.00002956 |

These figures are fairly accordant, and the loss at each draining-tank may be taken as

$$\mathfrak{H} = 0.00003pv^2. \quad . \quad . \quad (11)$$

*Experiments by Professor Riedler and Professor Gutermuth on the Resistance of the Paris Air-Mains.*—The old air-mains of Paris consist of cast-iron pipes 30 centimetres = 11¾ inches = 0.98 foot in diameter. It was on these that the experiments were made. For the new station wrought-iron pipes of 20 inches diameter are being used. The importance of these experiments lies chiefly in the large scale on which they were carried out. The volumetric efficiency of the compressors was first determined. Then, at different points on the pipe-line, the pressure was observed when the compressors were running at known speeds. On Sundays the whole pipe-line could be coupled up, all work in the city being stopped. The air then passed through the southern main and back by the northern main, a distance of 10 miles.

The main is somewhat complicated, having been constructed piecemeal during the gradual extension of the enterprise. In the 10-mile length there are four draining-tanks, twenty-three draining-traps or siphons, and forty-two stop-valves. The resistance of these is included in the observed results. It appears, however, that the most serious additional resistance was that of the draining-tanks, and as special experiments were made on these, their resistance can be estimated with very approximate accuracy and allowed for.

The following table gives the results of the experiments reduced to English measures. The columns marked with an asterisk are those taken directly from Professor Riedler's Tables. The others are deduced from his figures.

The last column gives the values of $\zeta$ calculated by equation (10).

It will be seen that the values of $\zeta$ are fairly consistent, considering the difficulties of the observations. The discrepancies are irregular, and obviously due to inconsistencies in the data of the experiments. For instance, experiments XIII and XIV were made on the same main at nearly the same initial pressure. In XIV the velocity was sensibly greater than in XIII, but the loss of pressure is much greater in XIII than in XIV.

Professor Riedler appears to believe that his experiments on the Paris main show that the friction of air in pipes is considerably less than it was believed to be from the results of earlier experiments on a smaller scale. The author does not think that this is so, but that, on the contrary, when properly reduced they are consistent with previous results, and that all the known experiments fairly support each other. In a paper on the Coefficient of Friction of Air flowing in long pipes,* it was shown that Mr. Stockalper's experiments gave the following results:

| | $\zeta =$ |
|---|---|
| Mean for 0.492-foot pipe...... | 0.00449 |
| " 0.656-foot pipe...... | 0.00377 |

---

* Minutes of Proceedings Inst. C.E., vol. lxiii. p. 299.

These experiments, as well as some others given in that paper, show that the value of $\zeta$ for air as well as for water decreases as the size of the pipe is larger. Though experiments on the flow of air are not numerous enough to furnish any very trustworthy law, the author gave in 1880 the following expression for the value of $\zeta$:

$$\zeta = 0.0027\left(1 + \frac{3}{10d'}\right).$$

This gives the following values for $\zeta$,

| | $\zeta =$ |
|---|---|
| Pipe 0.492 foot diameter… | ..0.00435 |
| " 0.656 " " | ......0.00393 |
| " 0.980 " " | ......0.00351 |

which are only a little different from those deduced from Stockalper's and Riedler's experiments.

Rounding off slightly the mean of Professor Riedler's results, it will be assumed in the remainder of this paper that, for air flowing in pipes of not less than 1 foot diameter, $\zeta = 0.003$. Putting this value in equations (7$a$) and (9),

$$u_1 = \sqrt{\left\{ 74{,}300{,}000 \frac{d'}{L} \frac{(p_1^2 - p_2^2)}{p_1^2} \right\}} \quad (12)$$

$$p_2 = p_1 \sqrt{\left\{ 1 - \frac{v_1^2 L}{74{,}300{,}000 d'} \right\}} . \quad . \quad (13)$$

*Loss of Pressure per Mile of Pipe.*—In order to indicate what kind of results this formula leads to, the following cases have been calculated:

1. Given the diameter of the pipe, and the initial pressure and velocity of the air entering the main, it is required to find the loss of pressure in 1 mile of transmission. By simple transposition, putting $L = 5280$,

$$p_2 = p_1 \sqrt{\left\{ 1 - \frac{v_1^2}{14{,}072 d'} \right\}} . \quad . \quad (14)$$

Assuming initial velocities of 25, 50, and 100 feet per second, and initial pressures of 50, 100, and 200 lbs. absolute, the following values are obtained:

| Diameter of Pipe. | Initial Velocity $v_1$. | Values of Terminal Pressure $p_2$, when | | | Percentage of Initial Pressure lost in One Mile. |
|---|---|---|---|---|---|
| | | $p_1 = 50$ | $p_1 = 100$ | $p_1 = 200$ | |
| Feet. | Feet per Second. | | | | |
| 1.0 | 25 | 48.8 | 97.7 | 195.4 | 2.4 |
| .. | 50 | 45.3 | 90.6 | 181.2 | 9.4 |
| .. | 100 | 26.9 | 53.8 | 107.6 | 46.2 |
| 2.0 | 25 | 49.4 | 98.9 | 197.8 | 1.2 |
| .. | 50 | 47.7 | 95.4 | 190.8 | 4.6 |
| .. | 100 | 40.1 | 80.3 | 160.6 | 19.8 |

The percentage of pressure lost in a mile is the same whatever the initial pressure. It must not, however, be assumed that the loss in 2 miles is double the loss in 1 mile. The velocity increases and the density diminishes going along the main.

It is clear that when the velocity initially in the main exceeds 50 feet per second, the loss of pressure becomes serious even in a distance of a mile. How far that involves a loss of efficiency will be considered later.

2. Another mode of looking at the

question is interesting. Suppose the percentage loss of pressure is assumed and the initial velocity calculated. In the following table this has been done for transmissions to distances of 1, 5, 10, and 20 miles.

| Length of Main in Miles. | Diameter of Main in Feet. | Initial Velocity $v_1$ when the percentage Loss of Pressure in Transmission is | | | | |
|---|---|---|---|---|---|---|
| | | 1 | 2½ | 5 | 10 | 20 |
| 1 | 1 | 16.7 | 26.3 | 37.0 | 52.5 | 73.1 |
| | 2 | 23.7 | 37.3 | 52.4 | 71.18 | 100.6 |
| 5 | 1 | 7.5 | 11.8 | 16.5 | 23.5 | 31.8 |
| | 2 | 10.6 | 16.6 | 23.3 | 33.2 | 44.9 |
| 10 | 1 | 5.3 | 8.3 | 11.3 | 16.6 | 22.5 |
| | 2 | 7.4 | 11.7 | 15.9 | 23.4 | 31.8 |
| 20 | 1 | 3.7 | 5.9 | 8.3 | 11.8 | 16.0 |
| | 2 | 5.4 | 8.3 | 11.7 | 16.6 | 22.6 |

Transmission is possible to the longest distances here assumed at velocities not impracticably low, with losses of pressure which would not hinder efficient use of the air.

## ACTION OF THE AIR-COMPRESSORS.

When an air-compressor is driven by a steam-engine there is a difference between the work done by the steam and the work done on the air, due to the work expended in friction of the mechanism and measured by the difference of area of the indicator-diagrams of the steam-cylinder and compression-cylinder. If the compressor is driven by water-power there will also be a corresponding loss in friction of the mechanism, probably not widely different in amount. It will be sufficient here to consider a compressor driven by steam.

Let U be the work expended, measured on the steam-cylinder indicator-diagram, and $U_1$ the corresponding work shown on the compressor-cylinder diagram. Then if $\eta_1$ is the efficiency of the mechanism,

$$U_1 = \eta_1 U.$$

It is convenient at present to take U and $U_1$ to be the work in foot-pounds per pound of air compressed. In Professor

Kennedy's tests of some of the older compressors at Paris it appeared that $\eta_1 = 0.845$. Experiments by Professor Gutermuth on the new Riedler compressor gave $\eta_1 = 0.87$, a result not widely different.

In all compressors there are some losses of work due to clearance, to imperfect action of the valves, to leakage, and to other causes. But unless the machine is badly constructed these need not be large. If the air is compressed to a pressure above that in the mains—as has happened in some cases—there is a loss, for the unbalanced expansion uselessly heats the air; but this again can be kept within narrow limits if the valves act promptly and properly. The chief loss of work in compressors is due to useless heating of the air. It is not, of course, impossible that some of the heat thus generated should be usefully employed. But to make use of it would involve complications. It is said that in Birmingham, where the distance of transmission is not great, some of the heat reaches the motors, and then there is an economy in the amount of air used. But practically in most cases heat given to the air in the compressor is in fact wasted before the air is used for motive purposes. A perfect compressor for power distribution would be one in which the air taken in at atmospheric pressure $p_a$ should be compressed isothermally to an absolute pressure $p_1$ equal to that in the mains at the compressing station, and then delivered into the mains without valve-resistance.

Such a machine working without friction, or clearance, or valve-losses, would require for each pound of air compressed an amount of work given by the equation

$$U_2 = P_a V_a \log_e \frac{p_1}{p_a} \text{ foot-lbs.},$$

where $P_a$ is the atmospheric pressure in lbs. per sq. ft., and $V_a$ the volume of 1 lb. of air at that pressure.

If, as hitherto assumed, the initial temperature of the air is 60° Fahr.,

$$U_2 = 27{,}690 \log_e \frac{p_1}{p_a}.$$

Fig. 2 shows the indicator-diagram of such an engine. OADE is the work

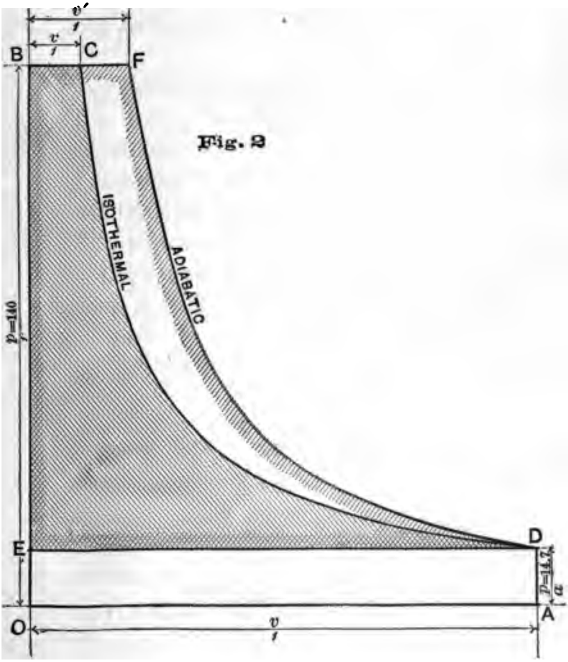

Fig. 2

Action of Compressor.

done by the atmosphere on the piston in the suction stroke. OBCDA is the work done on the air in the compressing stroke, the compression-curve DC being a common hyperbola. The work expended in compression is therefore the shaded area, EBCD.

If the air were compressed without any cooling during compression, the compression-curve would be an adiabatic, such as DF. The work expended in compression would be the area BFDE. Consequently in that case the area CFD would represent the work expended in useless heating of the air. This is easily calculated, for it can be shown that the work expended per pound of air in adiabatic compression is

$$\frac{\gamma}{\gamma - 1} \mathrm{P}_a \mathrm{V}_a \left[ \left( \frac{p_1}{p_a} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right].$$

Putting $\gamma = 1.408$ and $\mathrm{P}_a \mathrm{V}_a = 27{,}690$ as before,

$$\mathrm{U}_2' = 95{,}458 \left[ \left( \frac{p_1}{p_a} \right)^{0.29} - 1 \right]. \quad (15)$$

If $\eta_2 = \mathrm{U}_2/\mathrm{U}_2'$, then $\eta_2$ may be termed the efficiency of the compressing process.

For simple adiabatic compression, neglecting clearance and friction losses,

$$\eta_2 = \frac{\log_e \dfrac{p_1}{p_a}}{3.44\left[\left(\dfrac{p_1}{p_a}\right)^{0.29} - 1\right]}.$$

This may be called the theoretical value of $\eta_2$.

| $p_1$ | $p_a$ | $\dfrac{p_1}{p_a}$ | Theoretical Value of $\eta_2$. |
|---|---|---|---|
| 44.1 | 14.7 | 3 | 0.85 |
| 73.5 | .... | 5 | 0.78 |
| 102.9 | .... | 7 | 0.74 |
| 147.0 | .... | 10 | 0.70 |

It is clear, therefore, that if the air is not cooled during compression the efficiency of the process decreases as the pressure to which the air is compressed is greater. Now in most cases the cooling arrangements are very imperfect, and the compression is nearly adiabatic. Consequently there has been reluctance to use high initial pressures, and this diminishes the facility for distributing power by compressed air. As the other losses besides that due to heating are serious, the whole efficiency of the compressor is considerably less than the values calculated above.

Fig. 3 shows an actual diagram from one of the best of the older Colladon compressors. The shaded area BCDE is the useful work of compression, $U_2$ corresponding to the area EBCD in Fig. 2. The thick line is the actual indicator-diagram. It will be seen that the actual diagram is 38.15 per cent larger than the isothermal diagram for the volume of air

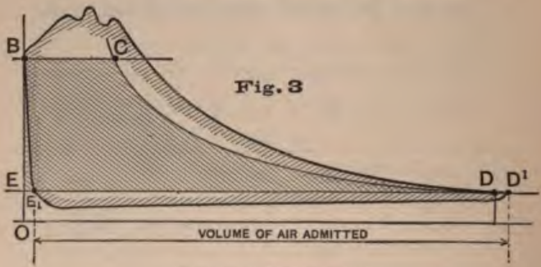

**Fig. 3**

COLLADON COMPRESSOR.

Revolutions, 104; work wasted, 38.15 per cent of useful work.

admitted, the difference being due partly to the waste of work in heating the air and partly to valve-resistance, causing a loss of pressure in the suction-stroke and an excess pressure during delivery into the mains. In this case $\eta_2$, measured from the diagram, is 0.72. In several compressors tried by Professor Riedler, the loss of work in the compressing process was twice as great as in this case. Fig. 4 shows a similar diagram for one of

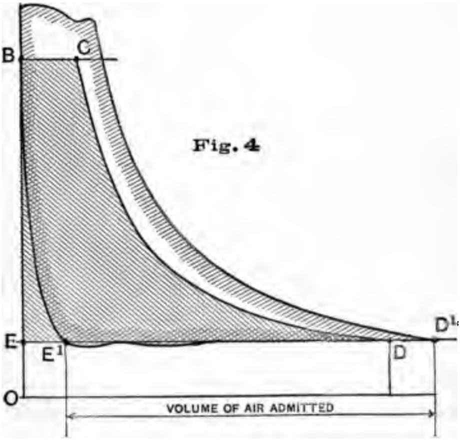

Fig. 4

Cockerill Compressor.

the Cockerill compressors at Paris. The excess work here is 40.2 per cent of the useful work, and consequently $\eta_2 = 0.70$.

Obviously, part of this loss can be saved if the air is cooled during compression, and all compressors are provided with some means of cooling. Very generally a water-jacket to the compressing cylinder is used, but the action of this is very imperfect. At Birmingham, although the compressors are well water-jacketed, and though the pressure is only 45 lbs. (gauge), the temperature of the air delivered is about 280°. The area of surface of the cylinder is small compared with the volume of air compressed, and air parts with heat to a metal surface slowly. Spray injection into the cylinder answers much better in keeping down the temperature. But it is believed that the result is partly deceptive, the cooling going on after the air is completely compressed, so that the compression-curve is hardly so much flattened as might be expected from the temperature at which the air is delivered.

The only remaining means of reducing the loss of work by heating is to compress

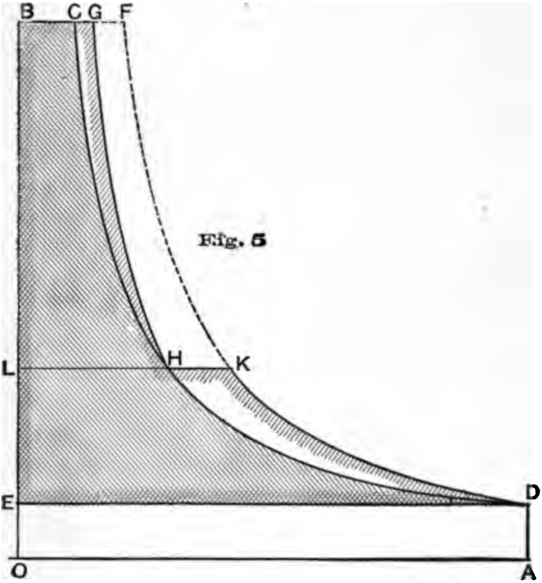

Fig. 5

ACTION OF COMPOUND COMPRESSOR.

in two or more stages, and cool the air thoroughly between the stages. This intermediate cooling can be easily effected, the air being taken through tubular vessels presenting any amount of cooling surface that may be required.

In Fig. 5, EBCD is the diagram of isothermal compression as before. DF is an adiabatic, so that in single-stage compression, without cooling, CDF represents the work lost in useless heating. Now let it be supposed that the compression is effected in two stages. In the first stage, no cooling being assumed, the work lost will be the area HKD. But the air is then cooled to its initial temperature, and the volume shrinks from LK to LH. H is a point on the isothermal DC. During the second stage HG is the adiabatic compression-curve, and CGH the work wasted in heating the air. It can now be seen that the effect of the intermediate cooling is to reduce the work expended in compression by the area GFKH a very material quantity.

It is this stage-compression with intermediate cooling, and with spray-injection also, which Professor Riedler has adopted in the new compressors for Paris, with marked increase of efficiency. Fig. 6 shows an actual diagram from one of the Riedler compressors, and it will be seen that the excess compression is only 12.07 per cent, so that in this case $\eta_2$ has the high value 0.89.

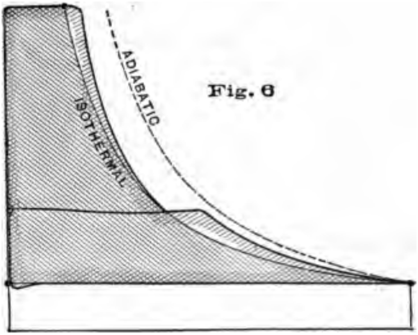

Fig. 6

RIEDLER COMPRESSOR.

Revolutions, 45; wasted work, 12.07 per cent of seful uwork.

In using air for transmitting power it is important, with a view of diminishing the size of the mains, to adopt high initial pressures, and it is the inefficiency of the compressing process in ordinary machines which has hitherto prevented the adoption of such pressures. Provided the efficiency of the compressors can be increased, much greater pressure loss can be permitted in the mains, and consequently much smaller mains will suffice to transmit a given amount of power.

## PROFESSOR RIEDLER AND PROFESSOR GUTERMUTH'S EXPERIMENTS ON THE EFFICIENCY OF COMPRESSORS.

If, as above, $U_1 = \eta_1 U$ is the work expended in compression after deducting the friction of the mechanism, and

$$U_2 = 27{,}690 \log_e \frac{p_1}{p_a}$$

is the useful work done in compression, then

$$\eta_2 = U_2/U_1 = U_2/\eta_1 U$$

is a coefficient of efficiency of the compressing process, which includes both the waste of work in useless heating and any losses due to clearance, valve resistance, etc.

Professor Riedler has obtained a series

of indicator-diagrams from different compressor cylinders. On these he has drawn an ideal isothermal diagram without clearance or valve-losses and for compression to the pressure in the main. The excess of area of the actual diagram over the ideal diagram is the work wasted in the compressing process, and from this $\eta_2$ is easily calculated. The following table gives a series of such results:

EFFICIENCY $\eta_2$ OF COMPRESSION.

| Type of Compressor. | Pressure in Main, $p_1$. Atmos. | Lost Work in Per Cent of Useful Work. | $\eta_2$ |
|---|---|---|---|
| Colladon, St. Gothard... | 6 | 105.0 | 0.488 |
| " " " ... | 6 | 92.0 | 0.521 |
| Sturgeon... .... .... .... | 3 | 94.3 | 0.515 |
| Colladon. ............... | 4 | 38.15 | 0.722 |
| Slide-valve compression. | 5 | 49.3 | 0.670 |
| Paxman........... ..... | ... | 42.7 | 0.701 |
| Cockerill ... ........... | 6 | 40.2 | 0.713 |
| Riedler two-stage........ | 6 | 12.07 | 0.892 |

The table shows how very low is the efficiency of some of the older compressors. Even in the compressors in Paris with single-stage compression $\eta_2 = 0.70$. Hence, if the friction of mechanism is taken account of, and $\eta_1$ is put at 0.85, the resultant efficiency is 0.85 × 0.70 = 0.595, or less than six-tenths of the indicated work in the steam-cylinder is usefully expended in compression. The Riedler two-stage compressor gives a much better result. Taking $\eta_1 = 0.87$, its efficiency is 0.77.

Professor Riedler has given some other figures, not based as the above are on the measurement of single diagrams, from which the efficiency of the Paris compressors can be calculated in a more trustworthy way. The following data are given as the result of a series of experiments with each compressor:

| Type of Compressors. | For each Indicated Steam H.P. | |
|---|---|---|
| | Cubic Feet of Air at Atmospheric Pressure compressed per Hour. | Pressure in Main. Lbs. per sq. inch Absolute. |
| Paxman machine (Sturgeon compressors) | 264.9 | 102.9 |
| Cockerill machine (Dubois François compressors) | 300.2 | 102.9 |
| Riedler machine (two-stage compressors) | 367.4 | 102.9 |

From these figures the following are calculated:

| | Steam Work per Hour in Ft.-lbs. | Weight of Air Compressed in Lbs. | Steam Work per Lb. of Air, = U. | Calculated Work per Lb. for Isothermal Compression, $U_2$. | $\eta_1 \eta_2 = \frac{U_2}{U}$. | $\eta_1$ | $\eta_2$ |
|---|---|---|---|---|---|---|---|
| Paxman... | 1,980,000 | 20.24 | 97,830 | 53,920 | 0.551 | 0.85 | 0.648 |
| Cockerill... | . ........ | 22.94 | 86,340 | ...... | 0.624 | 0.85 | 0.735 |
| Riedler.... | .... .... | 28.06 | 70,550 | ...... | 0.764 | 0.87 | 0.898 |

These results agree well with those obtained above in a different way.

## ACTION OF THE AIR-MOTORS.

An air-motor is simply a reversed air-compressor. Hitherto the conditions of efficiency in air-motors have received very little attention. In Paris many of those used are of small size, and in these a good efficiency is not to be expected. The best results have been obtained thus far by adapting old steam-engines to work as air-motors, and this can be done at very little trouble. It is specially desirable that in an air-motor the cylinder-clearance should be small or the compression sufficient. Probably the greatest source of avoidable waste in air-motors has been leakage at the piston. In a steam-engine, the condensation on the cylinder-wall helps to render the piston tight. In an ordinary air-engine the cylinder-surface is more or less dry, and the waste with air from leakage at even small apertures is very great.

A considerable economy can be secured by reheating the air in a simple form of stove before admitting it to the engine. At first sight this seems a complication likely to involve as much trouble as a steam-boiler. That, however, is not at all the case. The reheating appliances are extremely simple, there is no risk of explosion, and the amount of heat which it is desirable to give to the air is insignificant compared with that required in raising steam. Professor Riedler tried an old 80-H. P. steam-engine in Paris which had been adapted to work as an air-motor, and which was actually giving

72 indicated H. P. with compressed air at 5½ atmospheres pressure. It was using about 31,000 cubic feet (reckoned at atmospheric pressure) or about 2376 pounds of air per hour. This air was heated to a temperature of about 300° Fahrenheit by the expenditure of only 15 pounds of coke per hour. On a favorable assumption a steam-engine working to the same power would have required ten times this consumption of fuel at least. Reheating the air has the practical advantage of raising the temperature of exhaust of the motor, and for the amount of heat supplied the economy realized in the weight of air used is surprising. The reason of this is that the heat supplied to the air is used nearly five times as efficiently as an equal amount of heat employed in generating steam.

In certain cases the air-motor cylinder has been jacketed by hot air. This increases again the amount of work obtained per pound of air used. It can hardly be considered a thermodynami-

Second. Let it be supposed that the air, arriving in the main with the temperature $T_2$ (absolute), is reheated to a temperature $T_3$ before being used in the motor. As it is reheated at constant pressure, the amount of heat to be given to each lb. of air is 183.45 $(T_3 - T_2)$ foot-lbs., or 0.2377 $(T_3 - T_2)$ thermal units. Then the work of adiabatic expansion from a pressure $p_2$ to atmospheric pressure $p_a$ will be

$$U_3' = \frac{\gamma}{\gamma - 1} P_2 V_2 \frac{T_3}{T_2}\left[1 - \left(\frac{p_a}{p_2}\right)^{\frac{\gamma - 1}{\gamma}}\right]$$

$$= 95{,}458 \frac{T_3}{T_2}\left[1 - \left(\frac{p_a}{p_2}\right)^{0.29}\right] . \; . \quad (17)$$

Generally the work shown by the indicator-diagram of any engine will be less than this, for the causes mentioned above, and if $\eta_3$ is the efficiency with which the motor uses the fluid, the indicated work will be

$$U_4' = \eta_3 U_3'.$$

There is no reason for expecting $\eta_3$ to be different in this case from what it was in the previous one, unless hot-air jacketing, or steam injection, is used. In that case $\eta_3$ will be larger, and may be taken to include the additional work due to heat supplied during the stroke.

There is yet one more source of loss in the motor, the friction of the mechanism. If $\eta_4$ is the efficiency of the mechanism, then

$$U_5 = \eta_4 U_4, \quad \text{or} \quad \eta_4 U_4',$$

is the effective or brake work of the motor per pound of air used.

*Experiments on Air-motors by Professor Riedler and Professor Gutermuth.*—In the following tables some of the experiments are quoted, together with the work per pound of air and the values of $\eta_3$ and $\eta_4$. When Professor Riedler does not give the indicated, but only the brake, H. P. of the motor, it will be assumed that $\eta_4 = 0.85$ in order to calculate a probable value of $\eta_3$.

It will be seen that in the older small motors the efficiency with which the fluid is used ranges from 0.37 to 0.44, which perhaps, considering the kind of motor, is a good result. In the later machines, arranged to work expansively, the efficiency ranges from 0.58 to 0.87, results remarkably good for motors so small. The coefficients are rather higher with reheated air, showing that the work done is increased even in a rather higher ratio than $T_3/T_2$.

Some other experiments on small motors may be passed over in order to consider some experiments on an old Farcot steam-engine, with Corliss valves, which had been converted for use as an air-motor.

This engine was nominally of 80 H. P., and worked at 72 indicated H. P. in the trials. In all cases the air was reheated before use to about 300° Fahrenheit. The cylinder was also jacketed by the hot air on its way to the cylinder-chest.

The efficiency is therefore 0.81, an extremely good result.

Of all the work obtainable by the expansion of one pound of air received *at* 95.5 lbs. per sq. in., and at a tempera-

### SMALL ROTATING AIR-MOTORS.

| Nominal Size. | Revolutions per Minute. | Brake H. P. | Temperature of the Air. |  | Initial Pressure, Lbs. per Square Inch, Absolute. | Air used in Cubic Feet per Brake H. P. per Hour. | Air used in Lbs. per Brake H. P. per Hour. | Effective Work in Ft.-lbs. of 1 Lb. of Air $= U_5$. | Calculated Work of 1 Lb. of Air Expanded Adiabatically $= U_3$. | $\frac{U_5}{U_3} = \eta_3\eta_4$. | Assumed $\eta_4$. | $\eta_3$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  |  |  | At Admission | At Exhaust. |  |  |  |  |  |  |  |  |
| H. P. |  |  | Fahr. | Fahr. | *Old Motors without Expansion.* |  |  |  |  |  |  |  |
| ½ | 234 | 0.506 | 59 | 0 | 73.5 | 2,330 | 177.7 | 11,140 | 35,670 | 0.312 | 0.85 | 0 368 |
| 1 | 187 | 1.08 | 59 | 5 | 73.5 | 1,946 | 148.5 | 13,340 | 35,670 | 0.374 | 0.85 | 0.441 |
| ½ | 241 | 0.610 | 122 | .. | 73.5 | 1,671 | 127.5 | 15,530 | 45,100 | 0.344 | 0.85 | 0.405 |
| 1 | 230 | 1.42 | 140 | 74 | 73.5 | 1,625 | 124.0 | 15,970 | 46,500 | 0.343 | 0.85 | 0.404 |
|  |  |  |  |  | *Newer Motors with Expansion.* |  |  |  |  |  |  |  |
| 1 | 221 | 1.24 | 60 | −33 | 73.5 | 1,470 | 112.2 | 17,650 | 35,670 | 0.495 | 0.85 | 0.583 |
| 1 | 207 | 1.18 | 59 | −44 | 73.5 | 1,286 | 98.2 | 20,170 | 35,670 | 0.565 | 0.85 | 0.665 |
| 2 | 200 | 2.30 | 68 | −54 | 73.5 | 1,060 | 80.9 | 24,470 | 36,210 | 0.676 | 0.85 | 0.796 |
| 1 | 210 | 1.20 | 158 | 38 | 73.5 | 961 | 73.3 | 27,020 | 42,380 | 0.638 | 0.85 | 0.751 |
| 2 | 235 | 2.72 | 158 | .. | 73.5 | 848 | 64.7 | 30,610 | 42,380 | 0.722 | 0.85 | 0.850 |
| 2 | 241 | 3.22 | 140 | 28 | 73.5 | 848 | 64.7 | 30,610 | 41,150 | 0.744 | 0.85 | 0.876 |

## TRIALS WITH AN 80 HORSE-POWER FARCOT STEAM-ENGINE USED AS AN AIR-MOTOR.

| Initial Pressure. Lbs. per Square Inch Absolute. | Indicated H. P. | Temperature of the Air. | | Air used in Cubic Feet per Hour. | | Lbs. of Air per Hour. | | Indicated Work per Lb. of Air $= U_4$. | Brake Work per Lb. of Air $= U_5$. | Work per Lb. of Air Expanded Adiabatically $= U_5'$. | $\frac{U_5}{U_4} = \eta_4$. | $\frac{U_4'}{U_5'} = \eta_3$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Admission. | Exhaust. | Per Indicated H. P. | Per Brake H. P. | Per Indicated H. P. | Per Brake H. P. | | | | | |
| 95.5 | 72.3 | 264 | 70 | 469 | 517 | 35.7 | 39.5 | 55,460 | 50,130 | 62,120 | 0.90 | 0.89 |
| 95.5 | 72.3 | 305 | 84 | 437 | 475 | 33.3 | 36.2 | 59,460 | 54,700 | 65,630 | 0.92 | 0.90 |
| 95.5 | 72.3 | 320 | 95 | 425 | 465 | 32.4 | 35.5 | 61,120 | 55,770 | 66,920 | 0.91 | 0.91 |
| 95.5 | 65.0 | 338 | 120 | 438 | 477 | 33.5 | 36.4 | 59,100 | 54,400 | 68,470 | 0.92 | 0.87 |
| | | | | | | | | | | Mean .. | 0.91 | 0.89 |

ture of 300° Fahrenheit, four fifths is obtained as effective work on the brake. However, part of this work recovered is borrowed from the hot air before admission to the cylinder and given back to it by the jacket. That the jacket considerably affected the working during expansion is shown by the temperature during exhaust. The following table gives the temperatures the air would have reached by adiabatic expansion, and the actual temperatures of the exhaust in the experiments above:

| Initial Temperature. | | Final Temperature for Adiabatic Expansion. | | Actual Final Temperature, Fahr. | Increase of Temperature of Exhaust due to Jacket. |
|---|---|---|---|---|---|
| Fahr. | Absolute. | Fahr. | Absolute. | | |
| 264 | 725 | – 75 | 386 | + 70 | 145 |
| 305 | 766 | – 53 | 408 | + 84 | 137 |
| 320 | 781 | – 45 | 416 | + 95 | 140 |
| 338 | 799 | – 35 | 426 | + 120 | 155 |

## PRACTICAL CALCULATIONS ON THE DISTRIBUTION OF POWER BY COMPRESSED AIR.

It will be convenient first to give a summary of the formulas required in settling a system of compressed-air distribution, and afterwards to discuss some special cases.

Let $P_a$ be the pressure in lbs. per sq. ft., $p_a$ the pressure in lbs. per sq. in., $V_a$ the volume of a lb. in cu. ft., $T_a$ the absolute temperature of air admitted from the atmosphere to the compressor.

$P_1$, $p_1$, $V_1$, $T_1$ the same quantities for air discharged from the compressor into the main.

$P_2$, $p_2$, $V_2$, $T_2$ the same quantities for air arriving at the point of consumption in the main.

$U =$ indicated work done by the steam on the piston of the compressor reckoned in H. P.

$U_1 = \eta_1 U =$ corresponding indicated work in the compressor cylinder.

$U_2 = \eta_2 U_1$ = useful work of compression, or work of isothermal compression from $p_a$ to $p_1$.

$U_3$ = available work of air arriving at the motor, or work of adiabatic expansion from $p_2$ to $p_a$.

$U_4 = \eta_3 U_3$ = indicated power of the air-motor.

$U_5 = \eta_4 U_4$ = effective or brake work of the air-motor.

Taking $P_a = 2116.3$, $p_a = 14.7$, $V_a = 13.09$, $T_a = 521°$.

$$P_a V_a = 27{,}690. \qquad (1)$$

If W lbs. of air are compressed per second by U horse-power in the steam-cylinder corresponding to $U_1$ indicated horse-power in the compressor-cylinder, then, allowing both for friction of mechanism and clearance, leakage, and other losses in compression,

$$550 \eta_1 \eta_2 U = 27{,}690 W \log_e \frac{p_1}{p_a},$$

$$W = \frac{\eta_1 \eta_2 U}{50.3 \log_e \frac{p_1}{p_a}}. \quad . \quad . \qquad (2)$$

When by conduction in the main the air is at its initial temperature $T_a$,

$$P_1V_1 = 27{,}690,$$
$$p_1V_1 = 192.3. \quad . \quad . \quad . \quad (3)$$

If $v_1$ is the initial velocity of the air in the main, the diameter of which in feet is $d$,

$$\frac{\pi}{4}d^2v_1 = WV_1 = 192.3W/p_1,$$

$$d = 15.64\sqrt{\frac{W}{p_1v_1}}. \quad . \quad . \quad (4)$$

The pressure falls in a main of length $l$ in feet from $p_1$ to $p_2$, the amount being given by the equation

$$\frac{p_2}{p_1} = \sqrt{\left\{1 - \frac{v_1{}^2l}{74{,}300{,}000d}\right\}} \quad . \quad . \quad (5)$$

The available work of the air arriving at the motor with the pressure $p_2$ at atmospheric temperature $T_a$ is, if it is used without reheating, in H.P.,

$$U_3 = \frac{95,458W}{550}\left[1 - \left(\frac{p_a}{p_2}\right)^{0.29}\right]$$

$$= 173.5W\left[1 - \left(\frac{p_a}{p_2}\right)^{0.29}\right]. \quad . \quad (6)$$

If reheated to $T_3$ before admission to the motor,

$$U_3' = 173.5W\frac{T_3}{T_a}\left[1 - \left(\frac{p_a}{p_2}\right)^{0.29}\right]. \quad (7)$$

Indicated work of motor

$$= U_4 = \eta_3 U_3. \quad . \quad . \quad . \quad . \quad (8)$$

Brake or effective work of motor

$$= U_5 = \eta_4 U_4. \quad . \quad . \quad . \quad . \quad (9)$$

*Values assumed for the Coefficients of Efficiency.*—It appears that the efficiency of the mechanism of compressors is from 0.85 to 0.87. In the following calculations it will be assumed that $\eta_1 = 0.85$. As to the efficiency of the process of compression, this varies greatly with the type of compressor. In some of the older single-stage compressors it is as low as 0.5. But taking the best of those tried by Professor Riedler and slightly rounding off the values,

For single-stage compressors, $\eta_2 = 0.7$.
" two-stage " $\eta_2 = 0.9$.

The loss in the main must be calculated for each special case.

For the air-motors, as it is not intended to discuss the use of air on a small scale, it will be assumed that $\eta_3 = 0.85$ and $\eta_4 = 0.9$. These values are slightly below those obtained in the experiments, and the results should therefore be such as are practically realizable in ordinary work.

Case I.—10,000 H. P. is to be transmitted a distance of 2 miles. Taking $\eta_1 = 0.85$ and $\eta_2 = 0.9$ (two-stage compression) the useful work of compression will be 7650 H. P. Now, let the air be compressed to 4, 8, and 12 atmospheres on the gauge, then the weight of air compressed will be as follows:

| Case I. | Initial Gauge Pressure. Atmospheres | Initial Pressure Lbs. per Square Inch. Absolute. | $\frac{p_1}{p_a}$ | Weight of Air Compressed in Lbs. per Second = W. |
|---|---|---|---|---|
| *a* ..... | 4 | 73.5 | 5 | 94.54 |
| *b*..... | 8 | 132.3 | 9 | 69.10 |
| *c*...... | 12 | 191.1 | 13 | 59.29 |

To ascertain suitable diameters for the main, let initial velocities of 30, 50, and 75 feet per second be assumed. For these velocities, and in the cases described, the diameters required will be as follows:

| Case I. | Initial Pressure. Lbs. per Square Inch. Absolute. | Initial Velocity in the Main. Feet per Second. | Diameter of Main. | |
|---|---|---|---|---|
| | | | In Feet. | In Inches. |
| *a*... .. | 73.5 | 30 | 3.23 | 39 |
| | | 50 | 2.50 | 30 |
| | | 75 | 2.05 | 25 |
| *b*..... | 132.3 | 30 | 2.06 | 25 |
| | | 50 | 1.60 | 19 |
| | | 75 | 1.31 | 16 |
| *c*...... | 191.1 | 30 | 1.59 | 19 |
| | | 50 | 1.23 | 15 |
| | | 75 | 1.01 | 12 |

None of these mains are impracticable in size, and for the higher pressures and velocities they are surprisingly small, considering that they are shown to be capable of transmitting 10,000 indicated steam H. P. a distance of 2 miles. It remains to examine whether the loss of pressure is serious.

| Case I. | Initial Pressure. Lbs. per Square Inch Absolute $p_1$. | Initial Velocity in the Main. Feet per Second. | Diameter of the Main in Feet. | Terminal Pressure in Lbs. per Square Inch $p_2$. | Loss of Pressure in per Cent of Initial Pressure. |
|---|---|---|---|---|---|
| *a* ...... | 73.5 | 30 | 3.23 | 72.0 | 2.0 |
| | | 50 | 2.50 | 68.0 | 7.4 |
| | | 75 | 2.05 | 58.2 | 21.0 |
| *b*....... | 132.3 | 30 | 2.06 | 129.4 | 2.0 |
| | | 50 | 1.60 | 116.5 | 11.9 |
| | | 75 | 1.31 | 91.1 | 31.1 |
| *c*....... | 191.1 | 30 | 1.59 | 183.4 | 4.0 |
| | | 50 | 1.23 | 161.7 | 15.3 |
| | | 75 | 1.01 | 115.4 | 39.6 |

None of these losses would render the utilization of the power impossible, and it is only at the highest pressures and velocities that the fall of pressure is serious.

Lastly, the power developed at the air-motor due to 10,000 indicated steam H. P. in the compressor has to be calculated. The air being delivered at a distance of 2 miles, and used cold, the H. P. obtained will be as follows:

| Case I. | Initial Pressure. Lbs. per Square Inch Absolute. | Terminal Pressure in the Main. Lbs. per Square Inch Absolute. | Initial Velocity in the Main. Feet per Second. | Indicated H. P. of Air-motor. | Brake or Effective H. P. of Air-motor. |
|---|---|---|---|---|---|
| *a*...... | 73.5 | 72.0 | 30 | 5,146 | 4,631 |
| | | 68.0 | 50 | 4,998 | 4,498 |
| | | 58.2 | 75 | 4,596 | 4,136 |
| *b*....... | 132.3 | 129.4 | 30 | 4,785 | 4,306 |
| | | 116.5 | 50 | 4,600 | 4,140 |
| | | 91.1 | 75 | 4,177 | 3,759 |
| *c*....... | 191.1 | 183.4 | 30 | 4,546 | 4,091 |
| | | 161.7 | 50 | 4,376 | 3,938 |
| | | 115.4 | 75 | 3,934 | 3,540 |

It is very striking how little the efficiency is affected by considerable changes in the initial pressure and velocity in the mains; with the exception of two cases, for 10,000 steam indicated H. P. expended at the compressor there is obtained from 4400 to 5100 indicated H. P. at the air-motor. That is to say, the efficiency of the whole arrangement, compressor, main, and air-motor, when the air is used cold, ranges from 44 to 51 per cent, and that with mains of quite moderate size.

If the advantage of reheating at the motor is resorted to, which in no case involves any great additional trouble, and which from the small amount of fuel required can often be carried out with very little additional expense, the H. P. obtained will be as follows: Let it be supposed that the air arriving at the motor at 60° Fahrenheit is reheated to 300° Fahrenheit, as in the case of the Farcot steam-engine, details of which are given above. Then

$$\frac{T_3}{T_a} = \frac{761}{521} = 1.46.$$

| Case I. | Initial Pressure. Lbs. per Square Inch. | Terminal Pressure. Lbs. per Square Inch. | Initial Velocity in the Main. Feet per Second. | Indicated H. P. of Motor. | Brake H. P. of Motor. |
|---|---|---|---|---|---|
| *a*...... | 73.5 | 72.0 | 30 | 7,513 | 6,761 |
| | | 68.0 | 50 | 7,297 | 6,567 |
| | | 58.2 | 75 | 6,710 | 6,039 |
| *b*...... | 132.8 | 129.4 | 30 | 6,986 | 6,287 |
| | | 116.5 | 50 | 6,716 | 6,044 |
| | | 91.1 | 75 | 6,098 | 5,488 |
| *c*...... | 191.1 | 183.4 | 30 | 6,637 | 5,973 |
| | | 161.7 | 50 | 6,388 | 5,749 |
| | | 115.4 | 75 | 5,743 | 5,168 |

Here again, excepting two cases, the efficiency reckoned on the indicated power is from 64 to 75 per cent, neglecting the cost of the fuel for reheating.

Case II.—A long-distance transmission may now be discussed. Suppose, as before, that 10,000 indicated H.P. is developed in the steam-cylinders of the compressors, and is to be transmitted a distance of 20 miles. Taking the same pressures as before, the calculations of the weight of air compressed need not be repeated. The initial velocities previously assumed will, however, be excessive, because the velocity in the main

| Case II. | Initial Pressure. Lbs. per Sq. In. Absolute. | Initial Velocity. Feet per Second. | Diameter of Main in Feet. | Diameter of Main in Inches. |
|---|---|---|---|---|
| *d*........ | 73.5 | 20 | 3.96 | 47.5 |
| | | 35 | 2.99 | 35.9 |
| | | 50 | 2.51 | 30.1 |
| *e*........ | 132.3 | 20 | 2.53 | 30.4 |
| | | 35 | 1.91 | 22.9 |
| | | 50 | 1.60 | 19.2 |
| *f*........ | 191.1 | 20 | 1.95 | 23.4 |
| | | 35 | 1.47 | 17.6 |
| | | 50 | 1.23 | 14.8 |

increases as the pressure falls, and in this case, as the fall will be considerable, the terminal velocities are much greater than the initial velocities. Initial velocities of 20, 35, and 50 feet per second will be assumed.*

None of these mains are impracticable in size. The following table gives the calculation of the pressure loss:

| Case II. | Initial Pressure. | Initial Velocity. | Diameter of Main in Feet. | Terminal Pressure. Lbs. per Sq. In. | Loss of Pressure in per cent of Initial Pressure. |
|---|---|---|---|---|---|
| *d*....... | 73.5 | 20 | 3.96 | 68.0 | 7.4 |
| | | 35 | 2.99 | 47.6 | 35.3 |
| | | 50 | 2.51 | Impossible | .... |
| *e*....... | 132.3 | 20 | 2.53 | 116.5 | 11.9 |
| | | 35 | 1.91 | 38.4 | 70.9 |
| | | 50 | 1.60 | Impossible | .... |
| *f*....... | 191.1 | 20 | 1.95 | 160.8 | 15.8 |
| | | 35 | 1.47 | Impossible | .... |
| | | 50 | 1.23 | Impossible | .... |

It turns out on calculation that some of the cases assumed are impossible. That is, the whole initial pressure is insufficient to give the assumed initial velocity. In one other case, the 1.91 foot main with a velocity of 35 feet, the loss of pressure in the main is impracticably large. There remain, however, four cases in which neither the size of main nor the loss of pressure in a transmission to a distance of 20 miles is such as to render the transmission impracticable. In three cases the loss is remarkably small, and the terminal pressure quite suitable for application. For instance, it appears that the air compressed by 10,000 H. P. to 132.3 lbs. per sq. in. can be transmitted to a distance of 20 miles in a 30-inch main with a loss of pressure of only 12 per cent.

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* See table on page 111.

The power delivered at a distance of 20 miles by air-motors using the compressed air can now be calculated. If the air is used cold the power obtained will be as follows:

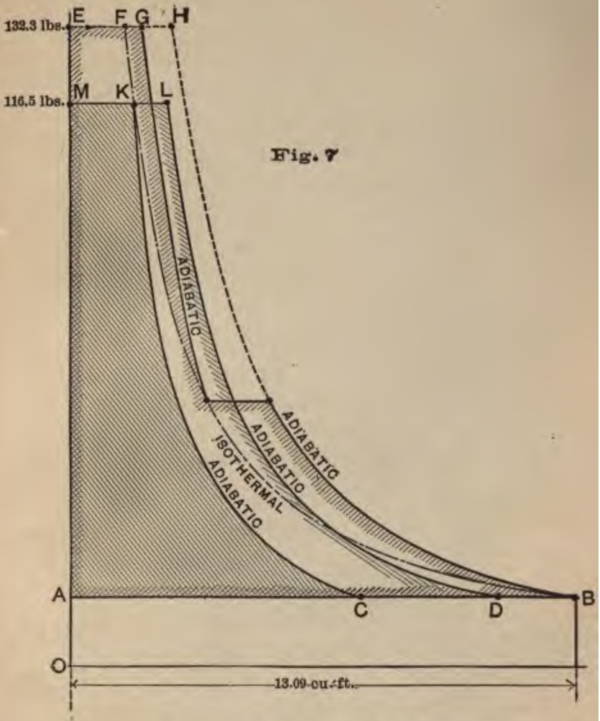
| Case II. | Initial Pressure. Lbs. per Sq. In. Absolute. | Terminal Pressure. Lbs. per Sq. In. Absolute. | Initial Velocity in Main. Feet per Second. | Indicated H. P. of Motor. | Brake or Effective H. P of Motor. |
|---|---|---|---|---|---|
| *d*....... | 73.5 | 68.0 | 20 | 4,989 | 4,490 |
| | | 47.6 | 35 | 4,017 | 3,615 |
| *e*....... | 132.3 | 116.5 | 20 | 4,598 | 4,138 |
| *f*....... | 191.1 | 160.8 | 20 | 4,365 | 3,928 |

If the air is reheated to 300° Fahr., as assumed in the previous case,

| Case II. | Initial Pressure. Lbs. per Sq. In. | Terminal Pressure. Lbs. per Sq. In. | Initial Velocity in Main. Feet per Second. | Indicated H. P. of Motor. | Brake or Effective H. P. of Motor. |
|---|---|---|---|---|---|
| *d*...... | 73.5 | 68.0 | 30 | 7,284 | 6,555 |
| | | 47.6 | 35 | 5,865 | 5,278 |
| *e*....... | 132.3 | 116.5 | 20 | 6,713 | 6,041 |
| *f*....... | 191.1 | 160.8 | 20 | 6,373 | 5,735 |

Here the efficiency of the whole arrangement—calculated on the indicated power, the air being delivered at a distance of 20 miles, and including all losses—is 40 to 50 per cent if the air is used cold, and 59 to 73 per cent if the air is reheated. The results are based absolutely on efficiencies already obtained in similar cases, and the sole loss neglected is possible leakage in the mains.

Fig. 7 is drawn to scale for Case II, *e*. It is a diagram showing the relation of the work expended and useful work recovered when 10,000 H. P. is transmitted 20 miles in a main 30 inches in diameter. AEGB is the work expended in a two-



Combined Action of Compressor Main and Motor.

stage compressor, compressing 1 lb. of air from 14.7 to 132.3 lbs. per sq. in. By cooling in the mains the volume of the air shrinks from EG to EF. The frictional resistance reduces the pressure to 116.5 lbs. per sq. in., when the air has the volume MK determined by the isothermal. The work done by an air-motor, using the air cold, is MKCA. If the air is reheated to 300° the volume expands from MK to ML (1.46 times). Then the work of the air-motor, using reheated air, is MLDA, and KLDC is the important gain of work due to reheating. Of course the areas given by the diagram have to be multiplied by the efficiencies given above.

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